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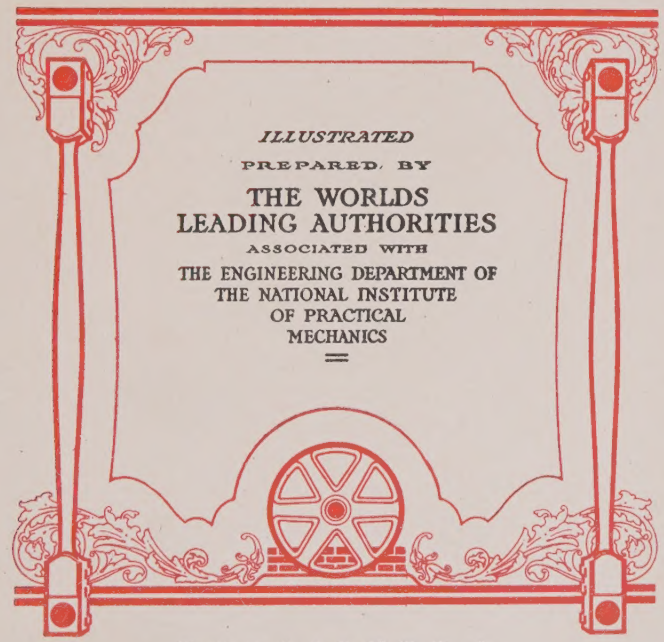

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MODERN AMERICAN ENGINEERING

A COMPLETE SERIES OF PRACTICAL TEXT BOOKS PREPARED ESPECIALLY FOR THE USE OF STEAM ENGINEERS, ELECTRICIANS, ERECTING ENGINEERS AND POWER USERS GENERALLY.

Prepared by a Corps of Experts, Electrical Engineers and Designers Connected With the National Institute of Practical Mechanics.

VOLUME IV

The Erecting Engineer, Planning and Arrangement of Factory Buildings, Strength of Material, Modulus of Elasticity, Mechanical Drawing and Machine Design, Shafting, Wire Rope Transmission, Belting, Gear Wheels, Electrical Transmission, The Water Turbine and the Principles Controlling Its Action, The Hydraulic Ram, Flow of Water, and Construction of Dams.

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PRACTICAL MECHANICAL DRAWING

DRAFTING TOOLS

Compasses—These, as well as other instruments, should be chosen with great care on account of their variety in shape and quality. Drafting instruments are as a rule made of German silver and steel. The steel should be of the best grade and carefully tempered. The material used in the manufacture of some instruments is of so poor a quality that it neither holds its

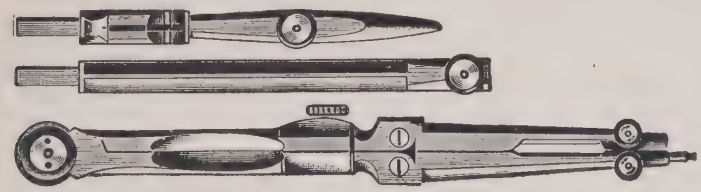


Fig. 1—Compass Set

shape nor wears well. It is better therefore when buying, to select instruments of high grade, and which are made of the best German silver and a good quality of steel, as the joints are always carefully fitted and they will withstand the constant usage of many years. A compass set of convenient form is shown in Fig. 1. It has three removable parts: the pencil-point, the pen-

point, and the lengthening bar. There is a hinged joint in each leg of the compass and the socket for the removable legs is provided with a clamping screw. The shanks of the removable legs should be a nice fit in the socket and require scarcely any effort to remove. They should, however, stay in the socket without being held by the clamping screw. The lengthening-bar is used to extend the pen or pencil legs when drawing large circles.

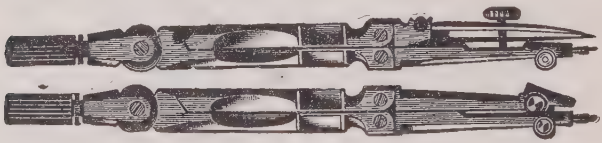


Fig. 2—Pen and Pencil Compasses

A pen and pencil compass set of smaller size, without detachable legs is shown in Fig. 2, these instruments will be found useful in many cases as a medium between the large compasses and the spring-bow set.

The most important part of a pair of compasses is the head, which forms the hinged joint. There are two forms of joints: the *tongue-joint*, as shown in the left-

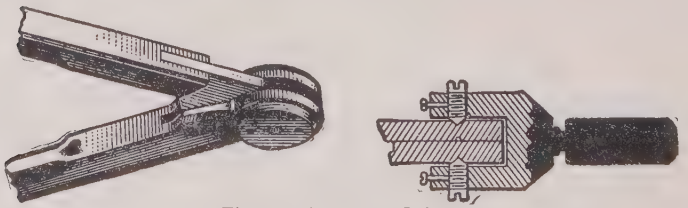


Fig. 3—Compass Joints

hand view in Fig. 3, in which the head of one shank has a tongue, generally made of steel, which moves between two lugs on the other shank, and the *pivot-joint*, as shown in the right-hand view in Fig. 3, in which

each shank is reduced to half its thickness at the head. These shanks are surrounded by a clamp or yoke, which carries two cone-pointed set-screws, one in each side, the points of these screws working in countersunks in the yoke. The yoke is provided with a milled or knurled handle to manipulate the compass. The head joint of the compass should move freely and evenly throughout its entire movement, and not stiff at one point and loose at another. It should also be tight enough in the joint to hold its adjustment when once set. Figure 4 shows

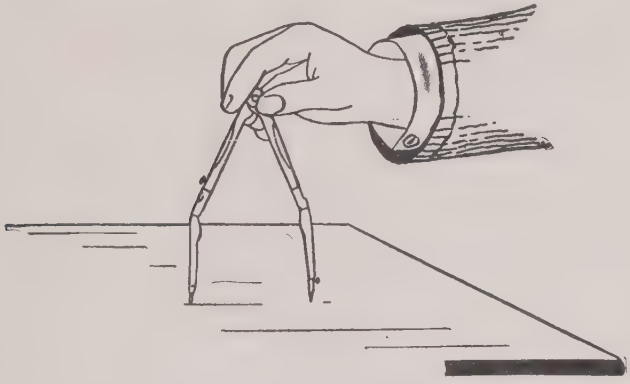


Fig. 4—Correct Manner of Holding a Compass

the method of holding a compass, and the correct position of the fingers before and after describing a circle. The non-removable leg of the compass should carry a needle-point, that may be easily replaced if lost or damaged, and it should have a shoulder to prevent the point from sinking into the paper beyond a certain depth. The needle-point should also be capable of being adjusted in or out, and fastened securely at any desired

point, thus making the leg of the compass a little longer or shorter as may be desired.

The socket for the lead in the pencil-point should hold the lead firmly without the necessity of wedging it in the socket by means of paper or small pieces of wood.

When first adjusting the compass for use, place the pen-point in the instrument and securely clamp it in

Fig. 5—Hair Spring Dividers

place, firmly against the shoulder of the socket, then adjust the needle-point so that its point is even with that of the pen. When once properly adjusted the needle-point should not be changed. The needle-point is usually made with a cone-point at one end and a fine shouldered-point at the other. The cone-point should never be used, as it makes too large a hole in the drawing paper.

Hair Spring Dividers—Dividers such as are shown in Fig. 5 are used for laying off equal distances and for transferring measurements from one part of a drawing to another, or from one drawing to another. They consist of steel points set in German silver shanks which are hinged together. The joints of the dividers should work smoothly, the legs come close together, and the steel points should be sharp and of the same length. One of the legs of the dividers has a spring controlled by an adjustable thumb-nut. By means of this device, from which the instrument gets its name, the adjustable

leg may be moved a trifle after the rough or approximate adjustment of the dividers has been made.

Spring-bow Instruments—The spring-bow dividers,

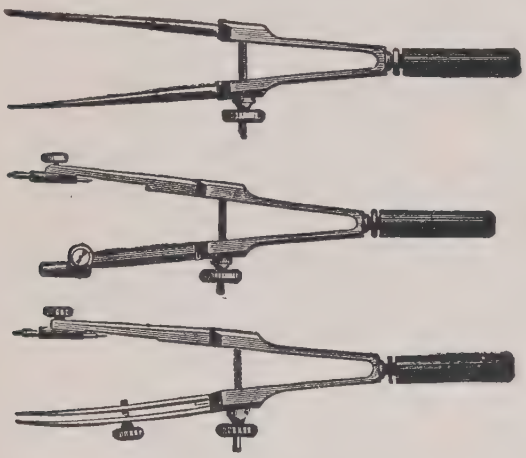


Fig. 6—Spring-bow Instruments

pencil and pen, as shown in Fig. 6, are for the purpose of describing small circles and laying off distances of very small dimensions and are very convenient for these

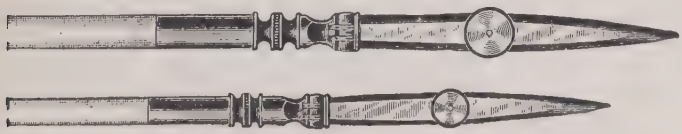


Fig. 7—Ruling Pens with Hinged Blade

purposes. Any form of spring-bow instruments with interchangeable or removable legs will be found very unsatisfactory. The legs should be made of one piece

of steel, to which the handle is attached. Any instrument in which the legs are separate pieces fastened to the shank are undesirable, because the parts are liable to become loose. The spring-bow dividers are used like the hair-spring dividers, for the spacing of distances, they have the advantage of being fixed in any position so that there is no liability of a change of measurement by the handling of the instrument. When spacing distances the divider is rotated alternately right and left, with the forefinger on top of the handle.

Ruling Pens—Ruling pens are of two different kinds, one kind with a hinge joint to allow the blades to be

Fig. 8—Ruling Pen with Spring Blade

opened for cleaning and the other kind without a joint and made from a solid piece of steel. Two sizes of ruling pens with hinged joints are shown in Fig. 7. The joint in this style of pen should be very carefully made, otherwise the hinged blade will very soon become loose and render the pen useless. The best kind of pen for general use is the kind shown in Fig. 8, in which the upper blade springs open when the adjusting screw is removed from the lower blade. A pen such as the one just described is to be preferred to one with a joint, no matter how well made it may be. Ruling pens with broad nibs and flat handles, as shown in Fig. 9, are preferred by many draftsmen, they hold a large quantity of ink and make a very uniform line.

The position in which a ruling pen should be held when drawing lines perpendicular to the T-square is shown in Fig. 10. The drawing board should be placed so as to permit the light to come from the upper left-

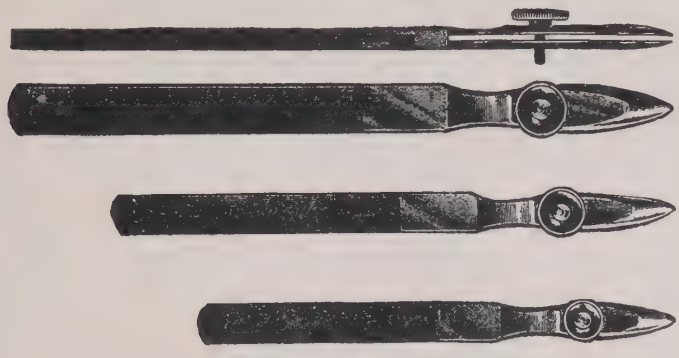


Fig. 9—Ruling Pens with Wide Blades

hand corner, this position of the T-square and triangle will avoid any possibility of the shadows of the T-square blade or triangle being cast on the lines to be drawn.

Sharpening a Ruling Pen—The blades of the pen

Fig. 10—Correct Manner of Holding a Ruling Pen

should be curved at the points, and elliptical in shape. To sharpen the pen, screw the blades together and then move the pen back and forth upon a fine oil-stone, holding it in the position it should have when in use, but moving it so that the points are ground to the same length, and to an elliptical form. When this form has been secured, draw a folded piece of the finest emery paper two or three times between the blades, which are pressed together by the screw. This will remove any roughnsss from the inner surfaces of the blades, these surfaces should not be ground upon the oil-stone.

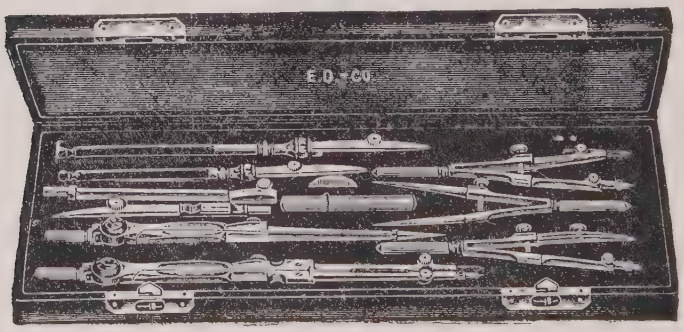


Fig. 11—Complete Set of Instruments in Case

When the blades are ground to the proper shape, they must be placed flat upon the stone and ground as thin as possible without giving them a cutting edge. To do this, the pen should be moved back and forth and slightly revolved at the same time. Both blades must be made of equal thickness. If either blade is ground too thin, it will cut the paper as would a knife, and the process must be repeated from the beginning. In order

to see the condition of the blades, they should be slightly separated while being brought to the proper thickness.

Drafting Instruments—A leather-covered case with a complete set of instruments in a velvet lined tray is shown in Fig. 11. This outfit is sufficient to fulfill the requirements of any ordinary draftsman in the way of instruments.

T-Square—The length of a T-square is always measured by the length of the blade outside of the head. The T-square should always be as long as the drawing-board, and if possible a little longer. For the general

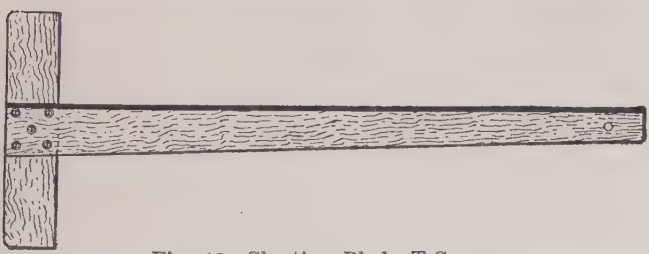


Fig. 12—Slanting Blade T-Square

run of work the head of the T-square should be of a single and fixed piece, that is, fastened permanently to the blade. The head should have its upper inside corner rabbeted, so that the guiding edge of the head may be trued up when occasion demands it. A very convenient form of T-square is shown in Fig. 12, which has a slanting blade, the working edge of which is lined with ebony.

More elaborate forms of T-squares are sometimes used, in which the head is double and one side swivels in order to draw parallel lines other than horizontal. The

adjustable or swivel head is clamped in any desired position by means of a thumb-screw.

Triangles or Set-Squares—Triangles are made of wood, hard rubber or transparent celluloid. The principal forms of triangles are shown in Figs. 13 and 14,

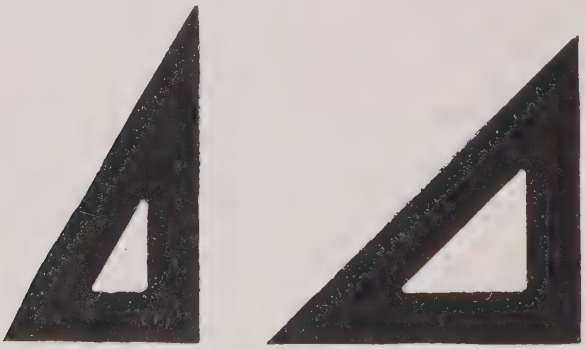


Fig. 13—60 and 45 Degree Triangles

which are 60°, 45° and 15° respectively. The two triangles generally used by draftsmen are the 60° and

Fig. 14—15 Degree Triangle

45°. The former has angles of 30°, 60° and 90°. The latter two 45° and a 90° angle.

Testing Triangles—Place the triangle on the T-square with the vertical edge at the right, draw a fine line in contact with this edge, then reverse the triangle and move the vertical edge towards the line. If the vertical edge of the triangle and the line coincide the angle is 90°. If they do not coincide, and the vertex of the angle formed by the line and the vertical edge of the triangle is at the top, the angle is greater than 90° by half the angle indicated. If the vertex of the angle is below, the angle is less than 90° by half the amount indicated.

Fig. 15—Flat Beveled-edge Scale

Scales—The best and most convenient form of scale for general use is that shown in Fig. 15. Another form of scale which is very commonly used in shown in Fig. 16. The ordinary length of a scale is 12 inches, not counting the small portion at each end, which is undivided, and whose use is to protect the end graduations

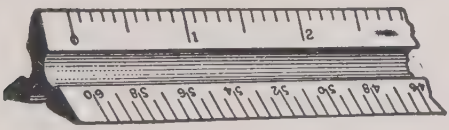


Fig. 16—Triangular Scale

from injury. A scale should be used for dimensioning drawings only, and not used as a ruler or straight-edge. The measurements should be taken directly from the scale by laying it on the drawing, and not by transferring the distances from the scale to the drawing by means of a pair of dividers.

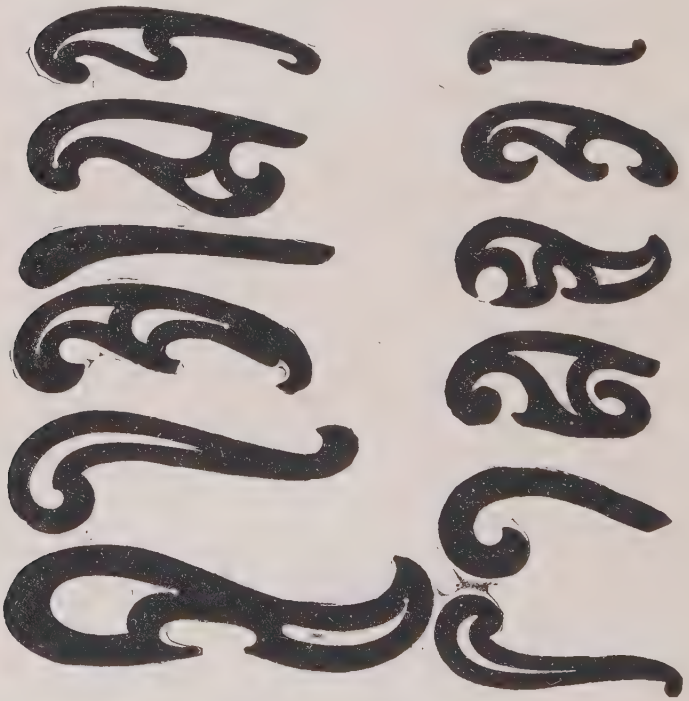


Fig. 17—French Curves

Curves—For inking in lines which are neither straight lines nor arcs of circles, it is necessary to use curves. They are made in a great variety of forms as illustrated

in Fig. 17, but the form similar to that illustrated in Fig. 18 will be found the most useful. They are made of wood, hard rubber and celluloid. Many curved lines can be inked in by means of a compass, but when the radius is too great, a curve should be used.

Paper—The paper must be tough and should have a surface which is not easily roughened by erasing lines drawn upon it. This is important when drawings are to be inked. For all mechanical work, the paper should be hard and strong.

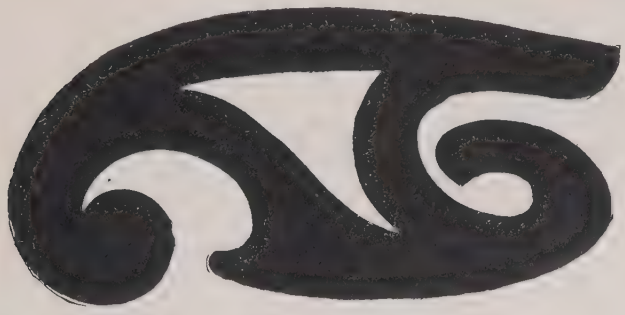


Fig. 18—Useful Form of Curve

For pencil drawings a paper which is not smoothly calendered is best, because the pencil marks more readily upon an unpolished paper, and because its surface will not show erasures as quickly as that of a smooth paper. For sketching, several kinds of paper, which are good enough for the work, may be obtained both in sheets, in block form, and also made up in blank books.

Whatman's paper is the best for drawings which are to be inked. There are two grades, hot and cold-pressed, suitable for this use, the cold-pressed having the rougher

surface. If the paper is not to be stretched, the cold-pressed is preferable, as its surface shows erasures less than that of the hot-pressed.

The side from which the water-marked name is read is the right side, but there is little difference between the two sides of hot and cold pressed papers. Stretching the paper is unnecessary except when colors are to be applied by the brush, or when the most perfect inked drawing is desired.

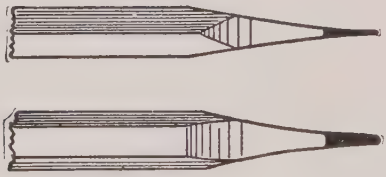


Fig. 19—Correct Manner of Sharpening a Pencil

Pencils—Lead pencils for drafting use are made of different degrees of hardness and each kind of pencil has its grade indicated by letters stamped on it at one end. The grade of pencil mostly used by draftsmen is 4 H; a 6 H pencil is too hard and unless used with great care will indent the paper so that the pencil marks cannot be erased. A 4 H pencil requires greater care and more frequent sharpening, but the draftsman will in this manner acquire a lighter touch, which is of much value. Drafting pencils should always be sharpened to a chisel or wedge-shaped point, as shown in Fig. 19, the finishing of the point should always be completed with a fine file or a sandpaper pencil sharpener, but never with a knife. In drawing the pencil should be held vertical, or nearly so, the arm free from the body,

and the flat side of the chisel-point lightly touching the edge of the blade of the T-square. Always draw from left to right, or from the bottom to the top of the board.

Fig. 20—Sand Paper Pencil Sharpener

Pencil Sharpeners—Pencil sharpeners or pointers are of many different kinds, from a piece of fine sandpaper or a file to quite complicated machines. For ordinary use a sandpaper block from which the sheets can be removed as soon as worn out will be found the most convenient, as shown in Fig. 20. In sharpening a drafting pencil remove the wood from the end by means of a sharp knife, exposing about one-fourth to three-eighths of an inch of the lead. The end of the lead should then be sharpened to a chisel or wedge-shaped point on the sandpaper block.

Fig. 21—Pencil Rubber or Eraser

Fig. 22—Ink Rubber or Eraser

Pencil Erasers—A pencil eraser or rubber should be of soft, fine-grained rubber ,free from sand or grit and

having no tendency to glaze or smear the surface of the drawing paper. A pencil eraser of the kind shown in Fig. 21 will be found very satisfactory for general use.

Ink Erasers—Inked lines should always be removed from the drawing by means of a sand-rubber, which is known as an ink eraser, but never by scratching the surface of the paper with a knife. As all drawing inks dry rapidly, and should not penetrate the surface of the paper, the object in erasing is to remove the ink from the

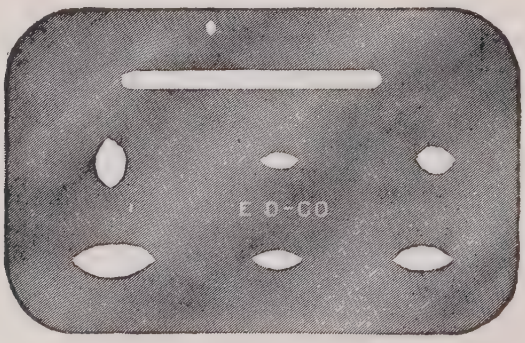


Fig. 23—Erasing Guard or Shield

surface of the paper without injury to it. An ink eraser, such as shown in Fig. 22, will leave the surface of the drawing in good condition to again receive ink.

Eraser Guard—An eraser guard or shield, which is used to protect other lines when removing an inked line from the surface of the drawing paper, consists of a thin sheet of flexible metal, usually brass, provided with slots and holes of various shapes and sizes. The shield or guard permits erasures to be made of limited size without damage to the rest of the drawing.

Drawing Ink—Black drawing ink, preferably some make of waterproof ink, is to be had in liquid form, as shown in Fig. 24. The liquid ink is preferable to the Indian or Chinese stick inks, shown in Fig. 25, which

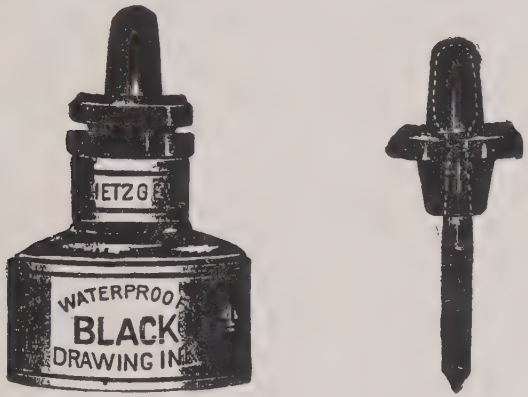


Fig. 24—Waterproof Black Drawing Ink

take considerable time to prepare, besides necessitating fresh mixing each time the ink is used.

Fig. 25—Chinese Stick Black Drawing Ink

Protractor—A protractor is a circular scale and is divided into degrees and fractions of a degree. Protractors are made both circular and semi-circular in shape, the latter being the ordinary and most commonly

used form, as shown in Fig. 26. Protractors are made of paper, horn, brass, German silver and steel. Protractors usually have their edges bevelled so as to bring

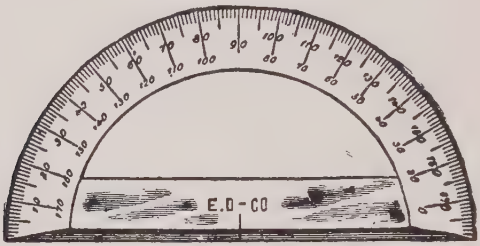


Fig. 26—German Silver Protractor

the divisions on the scale close to the drawing paper. A semi-circular protractor is to be preferred for all ordinary work. A semi-circular protractor has a straight edge upon which the center of the circle is marked, so that the protractor may be readily applied to the point at which it is desired to read or lay off an angle.

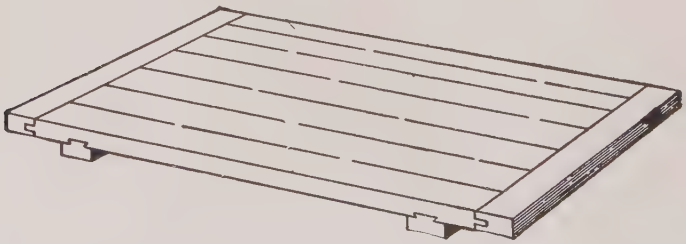


Fig. 27—Drawing Board with Cleats on Back

Drawing Boards—A drawing board for ordinary use should be about 20 by 27 inches in size. The material

should be of first quality clear soft pine, free from pitch and thoroughly kiln dried. The board should be made of five or six strips about 4 by 27 inches, well glued together and held from warping by two cleats on the back, as shown in Fig. 27.

The working edge of the drawing board should be tested from time to time, as any unevenness in this edge will impair the accuracy of the drawing. Some draftsmen use the lower edge of the board when drawing long lines parallel to the working edge. This necessitates making this edge true, and the angle between this and the working edge exactly 90°.

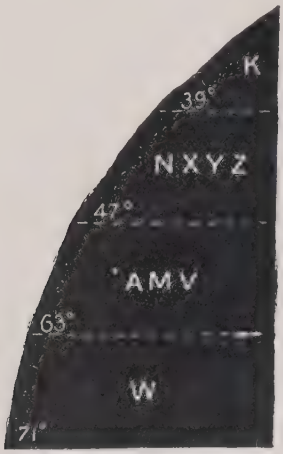


Fig. 28—Lettering Triangle

Thumb-Tacks—Thumb-tacks are made of German silver or brass disks with pointed steel pins in their centers. The heads or disks should have very thin edges

in order that the T-square may readily slide over them.

Lettering Triangle—A triangle or set-square for laying out lettering is shown in Fig. 28. The use of this triangle is plainly indicated by its name.

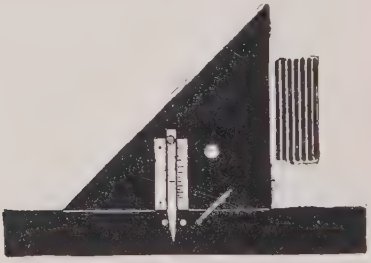


Fig. 29—Section Liner with Vertical Adjustment

Section Liners—A section liner is a device for drawing a series of parellel lines equi-distant from each other. One form of section liner is shown in Fig. 29. Its operation is as follows: Place the instrument in the

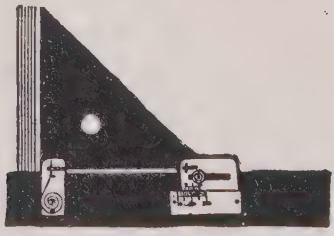


Fig. 30—Section Liner with Horizontal Adjustment

position shown in the drawing, and rule a line along its vertical edge. Hold the straight-edge firmly in place, and slide the triangle along it until the other side of the tapered edge of the tongue comes in contact with the other stud and holds it in this position, then allow the

straight-edge to be drawn forward by the spring. Then draw a second line which will evidently be parallel to the first. The distance between the lines is regulated by moving the tongue in or out between the studs, as far as desired.

Another form of section liner is shown in Fig. 30, having a horizontal instead of a vertical adjustment to regulate the width of the spacing.

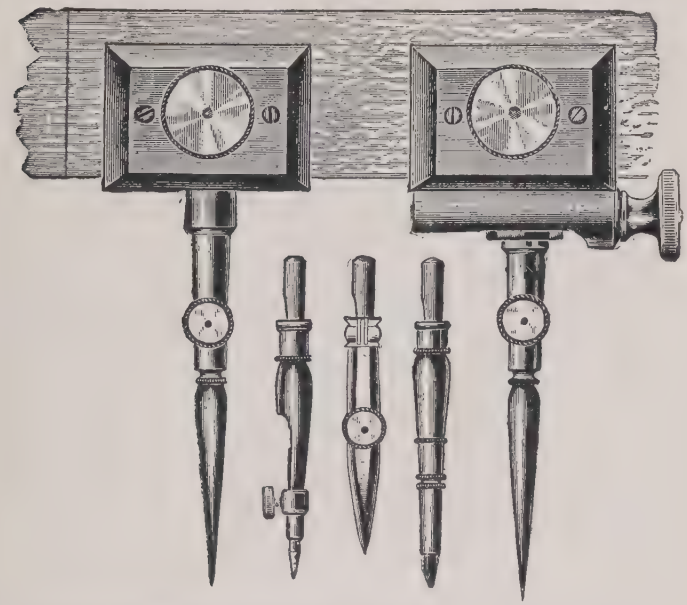


Fig. 31—Beam Compass with Pen, Pencil and Needle-Points

Beam Compasses—A beam compass is not, as a rule, included in a draftsman's outfit, but every well equipped drafting room should have one. A beam compass is

shown in Fig. 31, with removable legs and pen, pencil and needle-points. The right-hand leg in the illustration has a horizontal adjustment of about one-half an inch, operated by the milled thumb-nut shown.

Water Colors—These may be obtained in the form of a thick paste in small porcelain pans, or in thin paste or semi-liquid form in collapsible tubes. The colors in tubes are liable to get hard, in which case they cannot be expelled from the tubes by pressure. The caps to the tubes also get stuck in place by the colors and are often removed with much difficulty. For the draftsman's purposes the moist colors in pans will be found the most satisfactory.

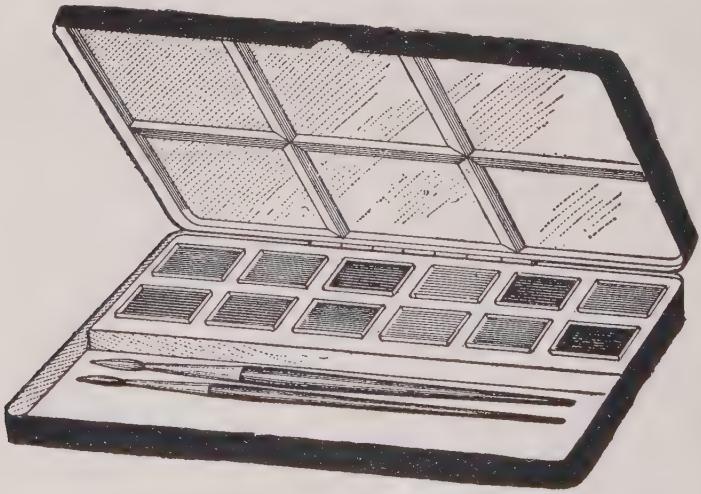


Fig. 32—Moist Water Colors in Case

A box of moist water colors is illustrated in Fig. 32. The colors should be kept in a box of this kind, which

can also be used as a palette. The box keeps all dust and dirt from the colors and prevents them from drying out rapidly.

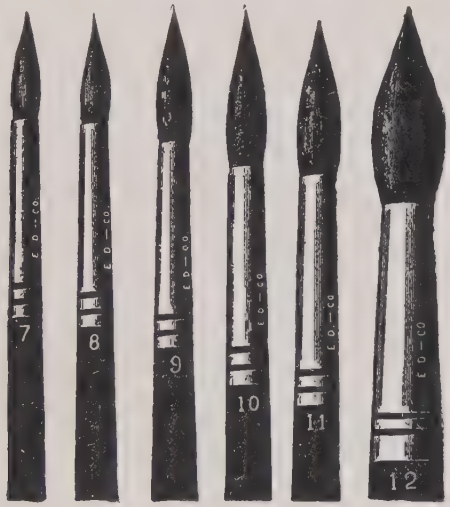


Fig. 33—Water Color Brushes

Water Color Brushes—These are made from black or red sable and camel's hair. Black sable brushes are too expensive for the draftsman's ordinary use.

The best grade of camel's hair brushes such as are shown in Fig. 33, will be found quite satisfactory for ordinary use.

To ascertain whether a brush is of good quality or not, dip it in water until thoroughly wet, and then remove the water from it by a quick motion. The brush if of good quality should assume a convex shape, come to a fine point and also preserve its elasticity.

GEOMETRICAL DEFINITIONS OF PLANE FIGURES

A line is the boundary or limit of a surface.

A line has only one dimension, that of length.

A point is considered as the extremity or limit of a line. The place where two lines intersect is also a point. A point has position but no dimensions.

In practical work a point is represented by a fine dot.

Lines may be either straight, broken or curved.

A straight line is one which has the same direction throughout its length.

A straight line is also called a *right line.*

A straight line is usually called a line simply, and when the word line occurs it is to be understood as meaning straight line unless otherwise specified. A straight line is the shortest distance between two points. If any other path between the points were chosen, the line would become curved or broken. Therefore two points determine the position of the straight line joining them.

A broken line is one which changes direction at one or more points.

A curved line is one which changes direction constantly throughout its length. The word curve is used to denote a curved line.

Lines may be represented as full, dotted, dashed, or dot-and-dashed.

A full line is one which is continuous throughout its length.

A dotted line is one which is composed of alternate dots and spaces.

A dash line is one which is composed of alternate dashes and spaces.

A dot-and-dash line is one which is composed of dots, spaces and dashes. These may be arranged in several ways according to the character of the line, that is, the meaning it is to convey.

Surfaces may be either plane or curved. A plane surface is usually called a plane.

A plane is such a surface that if a straight line be applied to it in any direction, the line and the surface will touch each other throughout their length.

A curved surface is one no part of which is a plane.

Any combination of points, lines, surfaces or solids is termed a *figure.*

A plane figure is one which has all of its points in the same plane.

Plane geometry treats of figures whose points all lie in the same plane.

Lines may be so situated as to be parallel or inclined to each other.

Parallel lines are those which have the same or opposite directions. Parallel lines are everywhere equally distant. Parallel lines will not meet, however far produced.

Inclined lines are those other than parallel. Inclined lines will always meet if produced far enough. Their mutual inclination forms an angle.

The *extremities of a surface* are lines.

A *plane rectilineal angle* is the inclination of two straight lines to one another in a plane which meet together, but are not in the same straight line as in Fig. 34.

When a straight line, standing on another straight line, makes the adjacent angles equal to one another,

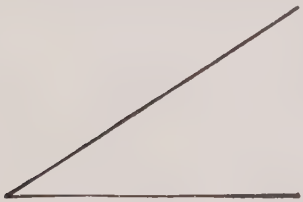


Fig. 34

Fig. 35

each of the angles is called a *right angle* and the straight line which stands on the other is called a *perpendicular* to it as in Fig 35.

Fig. 36

An *obtuse angle* is that which is greater than a right angle as in Fig. 36.

An *acute angle* is that which is less than a right angle as in Fig. 34.

A *term or boundary* is the extremity of anything.

An *equilateral triangle* is that which has three equal sides as in Fig. 37.

An *isosceles triangle* is that which has two sides equal as in Fig. 38.

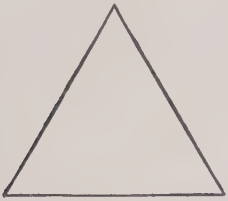


Fig. 37

Fig. 38

A *scalene triangle* is that which has three unequal sides as in Fig. 39.

A *right angled triangle* is that which has a right angle as in Fig. 40.

An *obtuse-angled triangle* is that which has an obtuse angle as in Fig. 39.

Fig. 39

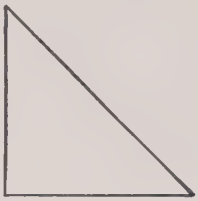


Fig. 40

The *hypothenuse* in a right angled triangle is the side opposite the right angle as in Fig. 40.

A *square* is that which has all its sides equal and all its angles right-angled as in Fig. 41.

A *rectangle* is that which has all its angles right angles, but only its opposite sides equal as in Fig. 42.

A *rhombus* is that which has all its sides equal, but its angles are not right angles as in Fig. 43.

Fig. 41

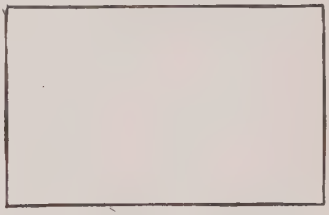


Fig. 42

A *quadrilateral* figure which has its opposite sides parallel is called a parallelogram as in Figs. 41, 42 and 43.

A line joining two opposite angles of a quadrilateral is called a *diagonal.*

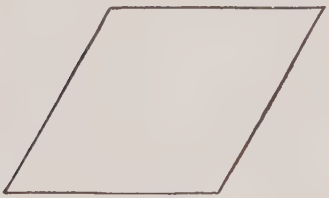


Fig. 43

Fig. 44

An *ellipse* is a plane figure bounded by one continuous curve described about two points, so that the sum of the distances from every point in the curve to the two foci may be always the same—Fig. 44.

PROPERTIES OF THE CIRCLE

A *circle* contains a greater area than any other plane figure bounded by the same length of circumference or outline.

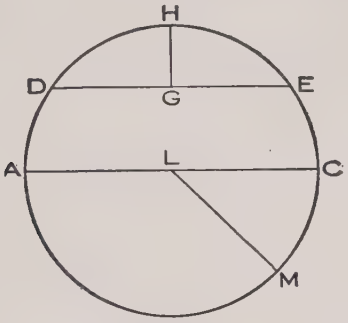


Fig. 45

A *circle* is a plane figure contained by one line and is such that all straight lines drawn from a point within the figure to the circumference are equal, and this point is called the center of the circle.

A *diameter* of a circle is a straight line drawn through the center and terminated both ways by the circumference, as AC in Fig. 45.

A *radius* is a straight line drawn from the center to the circumference, as LM in Fig. 45.

A *semicircle* is the figure contained by a diameter and that part of circumference cut off by a diameter as AHC in Fig. 45.

A *segment* of a circle is the figure contained by a straight line and the circumference which it cuts off, as DHE in Fig. 45.

A *sector* of a circle is the figure contained by two straight lines drawn from the center and the circumference between them, as LMC in Fig. 45.

A *chord* is a straight line, shorter than the diameter, lying within the circle, and terminated at both ends by the circumference as DE in Fig. 45.

An *arc* of a circle is any part of the circumference as DHE in Fig. 45.

The *versed sine* is a perpendicular joining the middle of the chord and circumference, as GH in Fig. 45.

Circumference—Multiply the diameter by 3.1416, the product is the circumference.

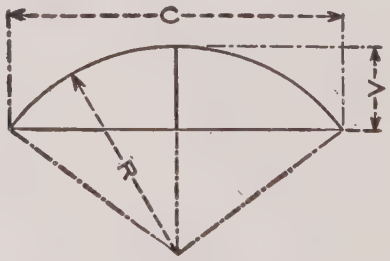


Fig. 46

Diameter—Multiply the circumference by .31831, the product is the diameter, or multiply the square root of the area by 1.12837, the produce is the diameter.

Area—Multiply the square of the diameter by .7854, the product is the area.

Side of the square—Multiply the diameter by .8862, the product is the side of a square of equal area.

Diameter of circle—Multiply the side of a square by 1.128, the product is the diameter of a circle of equal area.

To find the versed sine, chord of an arc or the radius when any two of the three factors are given—Fig. 46.

$$R=\frac{C^2+4V^2}{8V} \qquad C=2\sqrt{V(2R-V)}$$

$$V=R-\frac{\sqrt{4R^2-C^2}}{4}$$

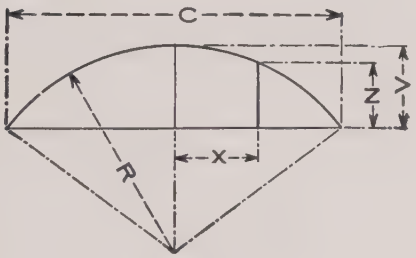


Fig. 47

To find the length of any line perpendicular to the chord of an arc, when the distance of the line from the center of the chord, the radius of the arc and the length of the versed sine are given—Fig. 47.

$$N=\sqrt{(R^2-X^2)}-(R-H) \qquad R=\frac{C^2+4V^2}{8V}$$

$$C=2\sqrt{V(2R-V)} \qquad V=R-\sqrt{\frac{4R^2-C^2}{4}}$$

To find the diameter of a circle when the chord and versed sine of the arc are given.

$$AC = \frac{DG^2 + GH^2}{GH}$$

Fig. 48

To find the length of any arc of a circle, when the chord of the whole arc and the chord of half the arc are given—Fig. 48.

$$\text{Arc } DHE = \frac{8DH - DE}{3}$$

A *Tangent* is a straight line which touches the circumference but does not intersect it. The point where the tangent touches the circle is called the *Point of Tangency.*

Two *Circumferences* are tangent to each other when they are tangent to a straight line at the same point.

A *Secant* is a straight line which intersects the circumference in two points.

A Polygon is *inscribed* in a circle when all of its sides are chords of the circle.

A Polygon is *circumscribed* about a circle when all of its sides are tangent to the circle, and a circle is circumscribed about a polygon when the circumference passes through all the vertices of the polygon.

DEFINITION OF POLYGONS

A *polygon,* if its sides are equal, is called a regular polygon, if unequal, an irregular polygon.

A *pentagon* is a five-sided figure.

A *hexagon* is a six-sided figure—Fig. 49.

Fig. 49—Hexagon

A *heptagon* is a seven-sided figure.

An *octagon* is an eight-sided figure.

A *nonagon* is a nine-sided figure.

A *decagon* is a ten-sided figure.

A *unadecagon* is an eleven-sided figure.

A *duodecagon* is a twelve-sided figure.

GEOMETRICAL DEFINITION OF SOLIDS

A *solid* has length, breadth and thickness. The boundaries of a solid are surfaces.

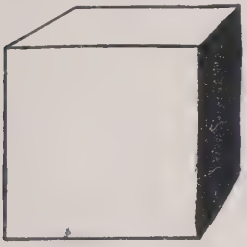


Fig. 50—Cube

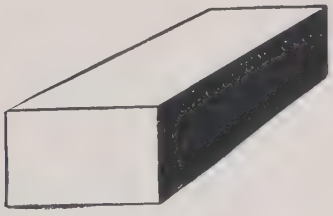


Fig. 51—Prism

A *solid angle* is that which is made by two or more plane angles, which are not in the same plane, meeting at one point.

A *cube* is a solid figure contained by six equal squares —Fig. 50.

A *prism* is a solid figure contained by plane figures of which two that are opposite are equal, similar, and parallel to one another, the other sides are parallelograms —Fig. 51.

A *pyramid* is a solid figure contained by planes, one of which is the base, and the remainder are triangles, whose verticles meet a point about the base, called the vertex or apex of the pyramid—Fig. 52.

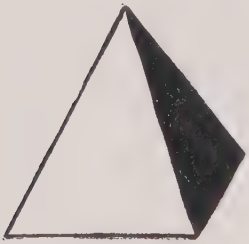


Fig. 52—Pyramid

Fig. 53—Cylinder

A *cylinder* is a solid figure described by the revolution of a rectangular or parallelogram about one of its sides—Fig. 53.

The *axis* of a cylinder is the fixed straight line about which the parallelogram revolves.

The *ends* of a cylinder are the circles described by the two revolving sides of the parallelogram.

A *sphere* is a solid figure described by the revolution of a semicircle about its diameter, which remains fixed—Fig. 54.

The *axis* of a sphere is the fixed straight line about which the semicircle revolves.

The *center* of a sphere is the same as that of the semicircle.

Fig. 54—Sphere

Fig. 55—Cone

The *diameter* of a sphere is any straight line which passes through the center and is terminated both ways by the surface of the sphere.

A *cone* is a solid figure described by the revolution of a right-angled triangle about one of its sides containing the right angle, which side remains fixed—Fig. 55.

The *axis* of a cone is the circle described by that side of the triangle containing the right angle which revolves.

The *base* of the cone is the circle described by that side of the triangle containing the right angle which revolves.

If a cone be cut obliquely so as to preserve the base entirely, the section is an *ellipse.*

When a cone is cut by a plane parallel to one of sloping sides, the section is a *parabola,* if cut at right angles to its base, an *hyperbola.*

MECHANICAL DRAWING

While many draftsmen are familiar with all of the problems given in this section of the work, it is not to be expected that all draftsmen or students are thoroughly conversant with all of them, and it is intended that this section of the work shall be used not only as reference data but for practical examples of elementary mechanical drawing. If the different problems given in this section are drawn with great accuracy, the technical skill acquired in drawing and proper handling of the different instruments will be found to be of great value.

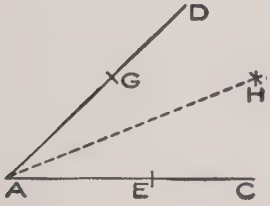


Fig. 56

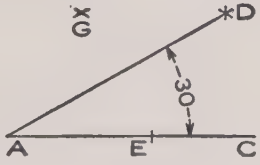


Fig. 57

It will not be necessary to ink in these simple geometrical problems, as it is better to acquire precision or accuracy in pencil work before going further. These problems are believed to be an essential part of a work on mechanical drawing. To understand geometry certain qualities of mind are absolutely necessary, and many persons find it impossible to grasp even the sim-

ple problems of this study. The draftsman or student who is without practical knowledge of geometry is very poorly equipped for his duties.

THE CONSTRUCTION OF ANGLES.

To bisect a given angle—Let DAC be the given angle. With center A and any radius AE describe an arc cutting AC and AD at E and G. With the same radius and centers E and G, describe arcs intersecting at H, and join AH. The angle DAC is bisected—Fig. 56.

To construct an angle of 30°—With radius AE and center A and E, describe arcs intersecting at G. With the same radius and with centers E and G, describe arcs intersecting at D, and join AD. The angle DAC contains 30°—Fig. 57.

To construct an angle of 60°—With radius AE, and with centers A and E, describe arcs intersecting at G, draw AD through G. The angle DAG contains 60°—Fig. 58.

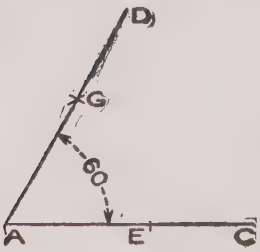


Fig. 58

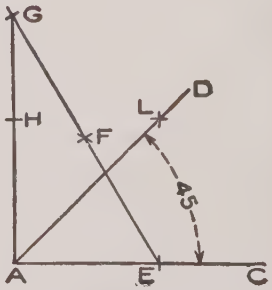


Fig. 59

To construct an angle of 45°—With radius AE and centers A and E, describe arcs intersecting at F, draw

EG through F, and make FG equal to FE. Join GR, and with center R and radius AE make AH equal to AE, with the same radius and with centers E and H describe arcs intersecting at L, draw AD through L. The angle DAC is 45°—Fig. 59.

To construct an angle of 90°—With radius AE and centers A and E, describe arcs intersecting at F, with

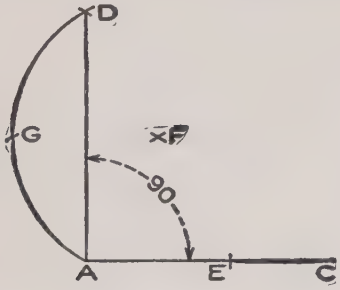


Fig. 60

the same radius and center F describe the arc AGD, with radius AE, lay off AG and GD and join DA. The angle DAG is 90°—Fig. 60.

To bisect a straight line—Fig. 61. Let BC be the straight line to be bisected. With any convenient radius greater than AB or AC describe arcs cutting each other at D and E. A line drawn through D and E will bisect or divide the line BC into two equal parts.

To erect a perpendicular line at or near the end of a straight line—Fig. 62. With any convenient radius and at any distance from the line AC, describe an arc of a circle as ACE, cutting the line at A and C. Through the center R of the circle draw the line ARE, cutting

the arc at point E. A line drawn from C to E will be the required perpendicular.

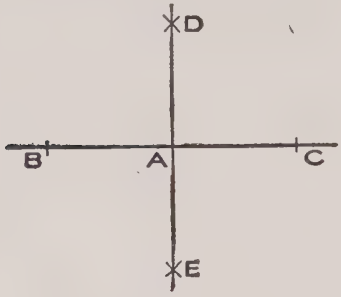


Fig. 61

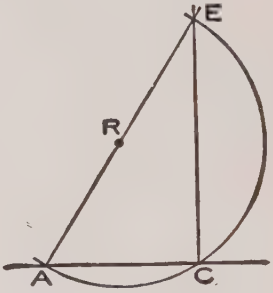


Fig. 62

To divide a straight line into any number of equal parts—Fig. 63. Let AB be the straight line to be divided into a certain number of equal parts: From the points A and B, draw two parallel lines AD and BC, at any convenient angle with the line AB. Upon AD and BC set off one less than the number of equal parts required, as A-1, 1-2, 2-D, etc. Join C-1, 2-2, 1-D, the line AB will then be divided into the required number of equal parts.

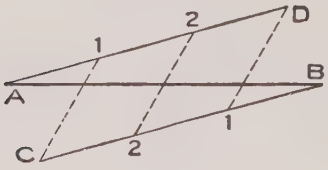


Fig. 63

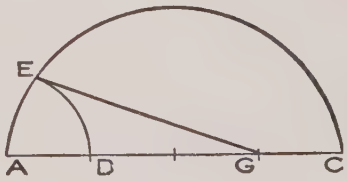


Fig. 64

To find the length of an arc of a circle—Fig. 64. Divide the chord AC of the arc into four equal parts as

shown. With the radius AD equal to one-fourth of the chord of the arc, and with A as the center, describe the arc DE. Draw the line EG and twice its length will be the length of the arc AEC.

To draw radial lines from the circumference of a circle when the center is inaccessible—Fig. 65. Divide

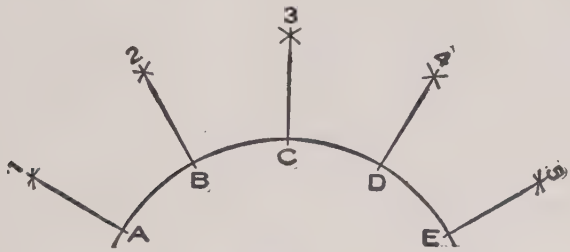


Fig. 65

the circumference into any desired number of parts as AB, BC, CD, DE. Then with a radius greater than the length of one part, describe arcs cutting each other as A-2, C-2, B-3, D-3, etc., also B-1, D-5. Describe the end arcs A-1, E-5 with a radius equal to B-2. Lines joining A-1, B-2, C-3, D-4 and E-5 will all be radial.

To inscribe any regular polygon in a circle—Fig. 66. Divide the diameter AB of the circle into as many equal parts as the polygon is to have sides. With the points A and B as centers and radius AB, describe arcs cutting each other at C. Draw the line CE through the second point of division of the diameter of AB, intersecting the circumference of the circle D. A line drawn from B to D is one of the sides of the polygon.

To cut a beam of the strongest shape from a circular section—Fig. 67. Divide any diameter CB of the circle

into three equal parts as CF, FE and EB. At E and F erect perpendiculars EA and FD on opposite sides of the diameter CB. Join AB, BD, DC and CF. The erect angle ABCD will be the required shape of the beam.

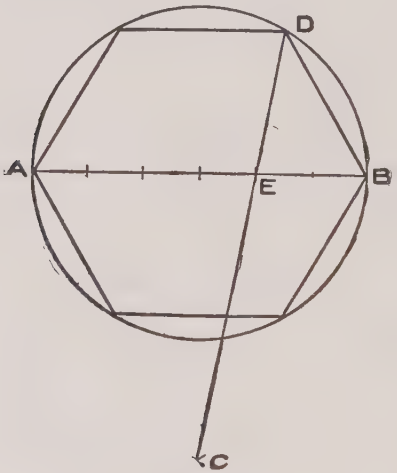


Fig. 66

To divide any triangle into two parts of equal area—Fig. 68. Let ABC be the given triangle: Bisect one of its sides AB at D and describe the semicircle AEB. At D erect the perpendicular DE and with center B and radius BE describe the arc EF which intersects the line AB at F. At F draw the line AG parallel at AC, this divides the triangle into two parts of equal area.

To inscribe a circle of the greatest possible diameter in a given triangle—Fig. 69. Bisect the angles A and B, and draw the lines, AD, BD, which intersect each other at D. From D draw the line CD perpendicular to AB.

Then CD will be the radius of the required circle CEF.

To construct a square equal in area to a given circle —Fig. 70. Let ACBD be the given circle: Draw the diameters AB and CD at right angles to each other, then bisect the half diameter or radius DB at E and draw the line FL, parallel to BA. At the points C and F erect the perpendiculars CH and FG, equal in length to CF. Join HG, then CFGH is the required square. The dotted line FL is equal to one-fourth the circle ACBD.

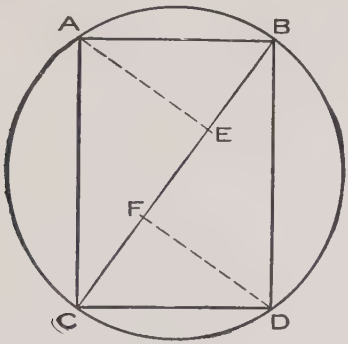


Fig. 67

To construct a rectangle of the greatest possible area in a given triangle—Fig. 71. Let ABC be the given triangle: Bisect the sides AB and BC at G and F. Draw the line GD and from the points G and D, draw the lines GF and DE perpendicular to GD, then EFGD is the required rectangle.

To construct a rectangle equal in area to a given triangle—Fig. 72. Let ABC be the given triangle: Bisect the base AB of the triangle at D and erect the perpen-

diculars DE and BF at D and B. Through C draw the line ECF intersecting the perpendiculars DE and B at E and F. Then BDEF is the required rectangle.

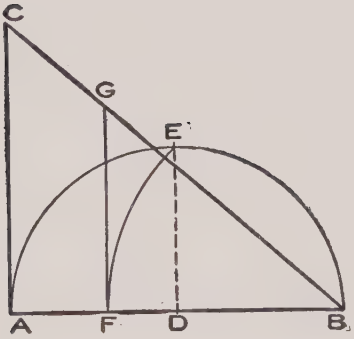


Fig. 68

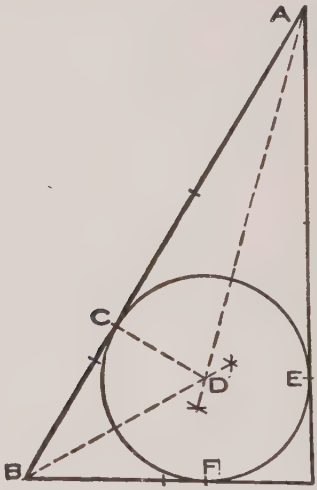


Fig. 69

To construct a triangle equal in area to a given parallelogram—Fig. 73. Let ABCD be the given parallelogram: Produce the line AB at B and make BE equal to AB. Join the points A and C and ACE will be the triangle required.

To inscribe a square within a given circle—Fig. 74. Let ADBC be the given circle: Draw the diameters AB and CD at right angles to each other. Join AD, DB and CA, then ACBD is the inscribed square.

To describe a square without a given circle—Fig. 75. Draw the diameters AB and CD at right angles to each other. Through A and B draw the lines EF and GH,

parallel to CD, also draw the lines EG and FH through the points C and D and parallel to AB, this completes the required square EFGH.

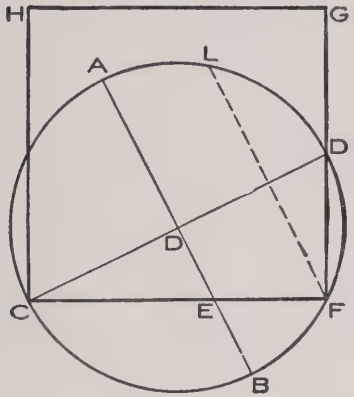


Fig. 70

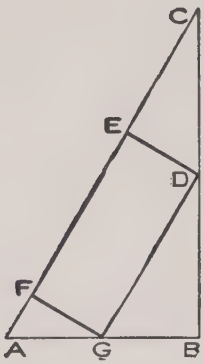


Fig. 71

To construct an octagon in a given square—Fig. 76. Let ABCD be the given square: Draw the diagonal lines AC and BD, which intersect each other at the

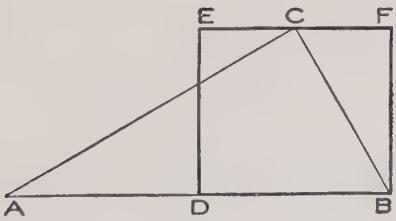


Fig. 72

point O. With a radius equal to AO or OC, describe the arcs EF, GH, IK, and LM. Connect the points EK,

LG, FI and HM, then GFIHMKEL is the required octagon.

To construct a circle equal in area to two given cir-

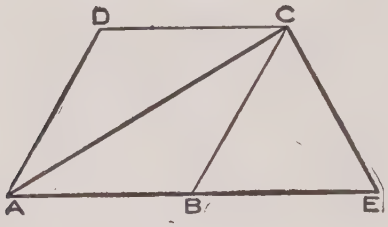


Fig. 73

cles—Fig. 77. Let AB and AC equal the diameters of the given circles: Erect AC at A and at right angles to AB. Connect B and C, then bisect the line BC at

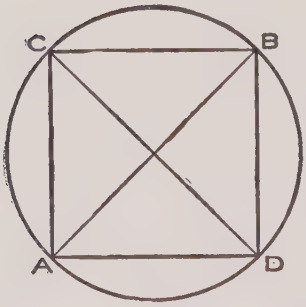


Fig. 74

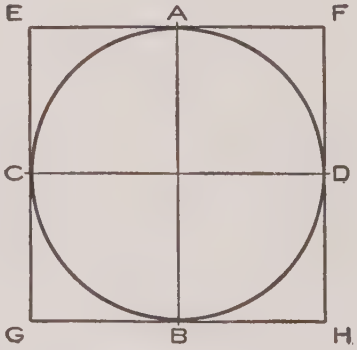


Fig. 75

D and describe the circle ACB which is the circle required and is equal in area to the two given circles.

To describe an octagon about a given circle—Fig. 78.

Let ACBD be the given circle: Draw the diameters AB and CD at right angles to each other. With any convenient radius and centers A, C, B and D describe arcs intersecting each other at E, H, F and G. Join EF and GH which form two additional diameters. At the points AB and CD draw the lines KL, PR, MN and ST parallel with the diameters CD and AB respectively. At the points of intersection of the circumference of the circle by the lines EF and GH, draw the lines KP,

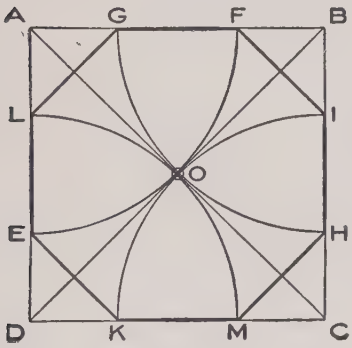


Fig. 76

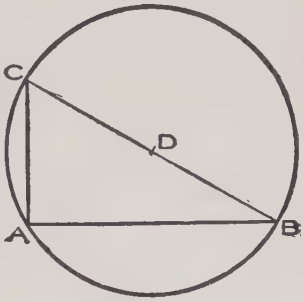


Fig. 77

RM, NT and SL parallel with the lines EF and HG respectively, then PRMNTSLK is the required octagon.

To draw a straight line equal in length to a given portion of the circumference of a circle—Fig. 79. Let ACBD be the given circle: Draw the diameters AB and CD at right angles with each other. Divide the radius RB into four equal parts. Produce the diameter AB and B and make BE equal to three of the four parts of RB. At A draw the line AF parallel to CD and then draw the line ECF which is equal to one-

fourth of the circumference of the circle ACBD. If lines be drawn from E through points in the circumference of the circle as 1 and 2, meeting the line AF and G and H, then C-1, 1-2 and 2-A will equal FG, GH and HA respectively.

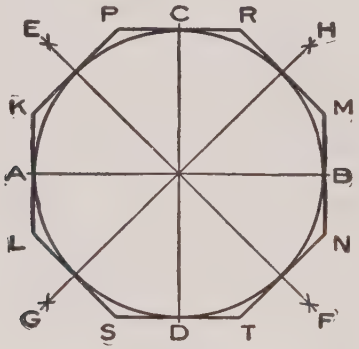


Fig. 78

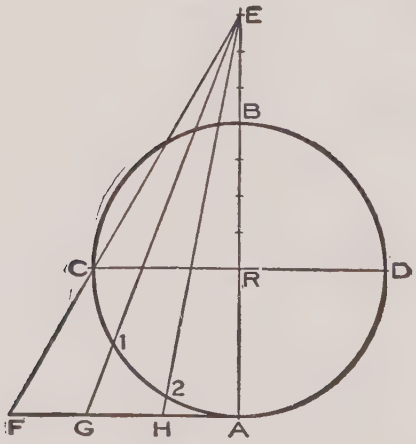


Fig. 79

To construct a square equal in area to two given squares—Fig. 80. Let AC and AD be the length of the sides of the given squares: Make AD perpendicular to AC and connect DC, then DC is one of the sides of the square DCEG which is equal to the two given squares.

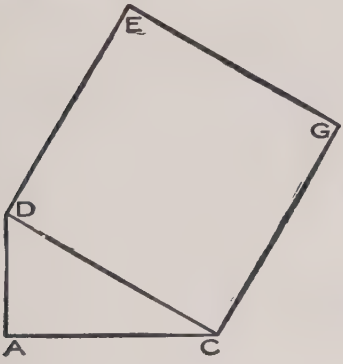


Fig. 80

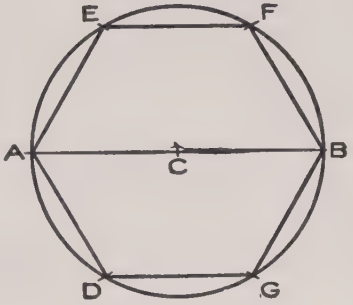


Fig. 81

To inscribe a hexagon in a given circle—Fig. 81. Draw a diameter of the circle as AB: With centers A and B and radius AC or BC, describe arcs cutting the circumference of the circle at D, E, F and G. Join

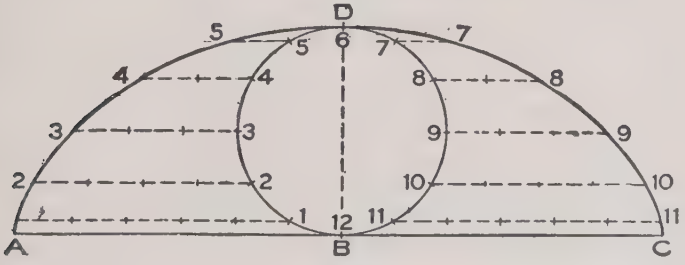


Fig. 82

EF, FB, BG, GD, DA and AE, this gives the required hexagon.

To describe a cycloid, the diameter of the generating circle being given—Fig. 82. Let BD be the generating circle: Draw the line ABC equal in length to the circumference of the generating circle. Divide the circumference of the generating circle into 12 parts as shown. Draw lines from the points of division 1, 2, 3, etc., of the circumference of the generating circle par-

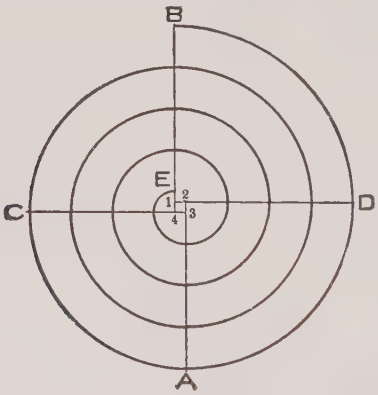


Fig. 83

allel to the line ABC and on both sides of the circle. Lay off one division of the generating circle on the lines 5 and 7, two divisions on the lines 4 and 8, three divisions on the lines 3 and 9, four divisions on the lines 2 and 10, and five divisions on the lines 1 and 11. A line traced through the points thus obtained will be the cycloid curve required.

To develop a spiral with uniform spacing—Fig. 83. Divide the line BE into as many equal parts as there are

required turns in the spiral. Then subdivide one of these spaces into four equal parts. Produce the line BE to 4, making the extension E-4 equal to two of the subdivisions. At 1 draw the line 1-D, lay off 1-2 equal to one of the subdivisions. At 2 draw 2-A perpendicular to 1-D and at 3 in 2-A draw 3-C, etc. With center 1 and radius 1-B describe the arc BD, with center 2 and radius 2-D describe the arc DA, with center 3 and radius 3-A, etc., until the spiral is completed. If carefully laid out the spiral should terminate at E as shown in the drawing.

MENSURATION

Mensuration is that branch of arithmetic which is used in ascertaining the extension and solidity or capacity of bodies capable of being measured.

DEFINITIONS OF ARITHMETICAL SIGNS.

=Sign of Equality, as 4+8=12.
+Sign of Addition, as 6+6=12, the Sum.
—Sign of Subtraction, as 6—3=3, the Remainder.
×Sign of Multiplication, as 8×4=32, the Product.
÷Sign of Division, as 24÷6=4 24-6=4.
√Sign of Square Root, signifies Evolution or Extraction of Square Root.
^{2} Sign of to be Squared, thus $8^2=8\times8=64$.
^{3} Sign of to be Cubed, thus $3^3=3\times3\times3=27$.

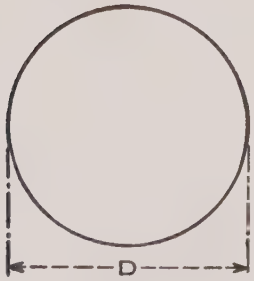


Fig. 84—Circle

Fig. 85—Semi-circle

MENSURATION OF PLANE SURFACES.

To find the area of a circle—Fig. 84. Multiply the square of the diameter by .7854.

To find the circumference of a circle. Multiply the diameter by 3.1416.

Circle: Area= $.7854D^2$
Circ.=3.1416D

To find the area of a semi-circle—Fig. 85. Multiply the square of the diameter by .3927.

To find the circumference of a semi-circle. Multiply the diameter by 2.5708.

Semi-Circle: Area= $.3927D^2$
Circ.=2.5708D

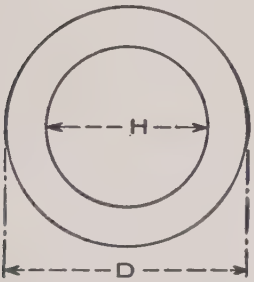


Fig. 86—Annular Ring

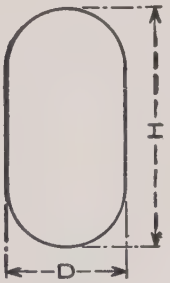


Fig. 87—Flat Oval

To find the area of an annular ring.—Fig. 86. From the area of the outer circle subtract the area of the inner circle, the result will be the area of the annular ring.

To find the outer circumference of an annular ring. Multiply the outer diameter by 3.1416.

To find the inner circumference of an annular ring. Multiply the inner diameter by 3.1416.

Annular ring: Area= .7854 (D^2—H^2)
Out. circ.=3.1416 D
Inn. circ.=3.1416 H

To find area of a flat-oval—Fig. 87. Multiply the length by the width and subtract .214 times the square of the width from the result.

To find the circumference of a flat-oval. The circumference of a flat-oval is equal to twice its length plus 1.142 times its width.

Flat-oval: Area=D (H—0.214D)
Circ.=2 (H×O.571D)

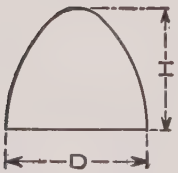


Fig. 88—Parabola

Fig. 89—Square

To find the area of a parabola—Fig. 88. Multiply the base by the height and by .667.

Parabola: Area=.667(D×H)

To find the area of a square—Fig. 89. Multiply the length by the width, or, in other words, the area is equal to square of the diameter.

To find the circumference of a square. The circumference of a square is equal to the sum of the lengths of the sides.

Square: Area=D^2
Circ.=4D

To find the area of a rectangle—Fig. 90. Multiply the length by the width, the result is the area of the rectangle.

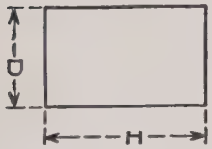


Fig. 90—Rectangle

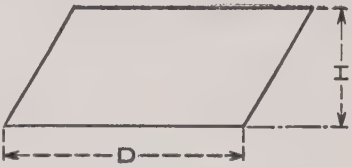


Fig. 91—Parallelogram

To find the circumference of a rectangle. The circumference of a rectangle is equal to twice the sum of the lenght and width.

Rectangle: Area=D×H
Circ.=2(D×H)

To find the area of a parallelogram—Fig. 91. Multiply the base by the perpendicular height.

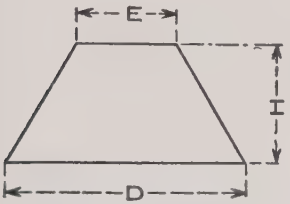


Fig. 92—Trapezoid

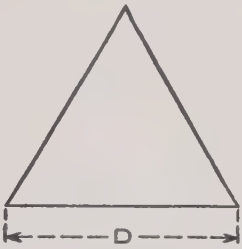


Fig. 93—Equilateral Triangle

Parallelogram: Area=D×H

To find the area of a trapezoid—Fig. 92. Multiply half the sum of the two parallel sides by the perpendicular distance between the sides.

Trapezoid: $\text{Area} = \frac{(HE+D)}{2}$

To find the area of an equilateral triangle—Fig. 93. The area of an equilateral triangle is equal to the square of one side multiplied by .433.

To find the circumference of an equilateral triangle. The circumference of an equilateral triangle is equal to the sum of the length of the sides.

Equilateral triangle: $\text{Area} = .433D^2$

$\text{Circ.} = 3D$

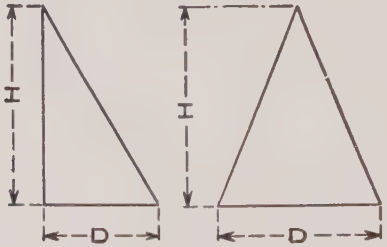


Fig. 94—Right-angle and Isosceles Triangles

To find the area of a right-angle or an isosceles triangle—Fig. 94. Multiply the base by half the perpendicular height.

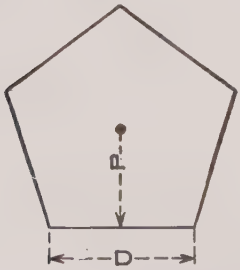


Fig. 95—Regular Polygon

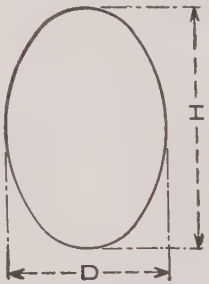


Fig. 96—Ellipse

To find the circumference of any regular polygon—Fig. 95. The circumference of any polygon is equal to the sum of the length of the sides.

Polygon: $\text{Area} = \dfrac{\text{No. of sides} \times D \times P}{2}$

Circ.=No. of sides×D

D=Length of one side.

P=Perpendicular distance from the center to one side.

To find the area of an ellipse—Fig. 96. Multiply the long diameter by the short diameter and by .7854.

To find the circumference of an ellipse. Multiply half the sum of the long and short diameters by 3.1416.

Ellipse: Area= .7854 (D×H)

Circ.=1.5708 (D+H)

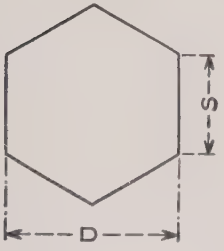


Fig. 97—Hexagon

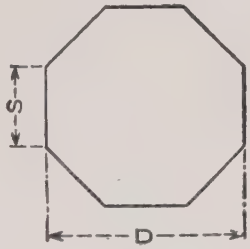


Fig. 98—Octagon

To find the area of a hexagon—Fig. 97. Multiply the square of one side by 2.598.

To find the circumference of a hexagon. The circumference of a hexagon is equal to the length of the sides.

Hexagon: Area=2.598 S^2
Circ.=6 S
D=1.732 S

To find the area of an octagon—Fig. 98. Multiply the square of the short diameter by .828.

To find the circumference of an octagon. The circumference of an octagon is equal to the sum of the length of the sides.

Octagon: Area=.828 D^2
Circ.=8 S
S=.414 D

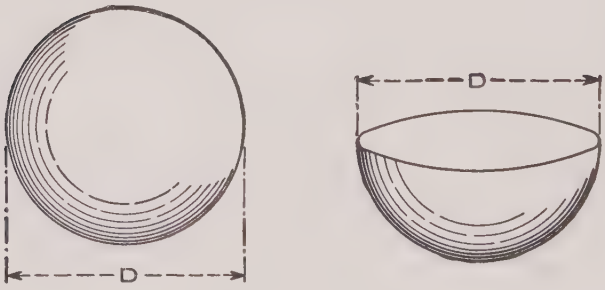


Fig. 99

To find the cubic contents of a sphere—Fig. 99. Multiply the cube of the diameter by .5236.

To find the superficial area of a sphere. Multiply the square of the diameter by 3.1416.

Sphere: Cubic contents=.5236D^3
Superficial area=3.1416D^2

The area of the surface of a sphere is equal to the area of the surface of a cylinder, the diameter and the height of which are each equal to the diameter of the

sphere. Also, the area of the surface of a sphere is equal to four times the area of its diameter.

The latter definition is easily remembered, and is useful in calculating the areas of the hemispheres, because the area of the sheet or disc of metal required for raising a hemisphere must be equal in area to the combined areas of two discs, each equal to the diameter of the hemisphere.

To find the cubic contents of a hemisphere—Fig. 99. Multiply the cube of the diameter by .2618.

To find the superficial area of a hemisphere.

Hemisphere: Cubic contents=.2618D^2
Superficial area=2.3562D^2

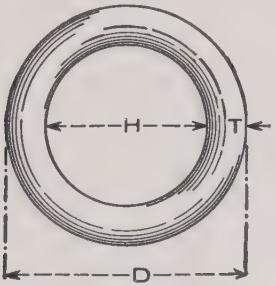


Fig. 100

To find the cubic contents of a cylindrical ring—Fig. 100. To the cross-sectional diameter of the ring add the inner diameter of the ring, multiply the sum by the square of the cross-sectional diameter of the ring and by 2.4674, the product is the cubic contents.

To find the superficial area of a cylindrical ring. To the cross-sectional diameter of the ring add the inner

diameter of the ring. Multiply the sum by the cross-sectional diameter of the ring and by 9.8696, the product is the superficial area.

Cylindrical ring: Cubic contents=$2.4674T^2$ (T+H)
Superficial area=9.8696T (T+H)
D=(H+2T)

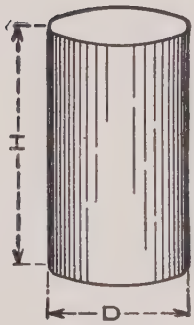


Fig. 101

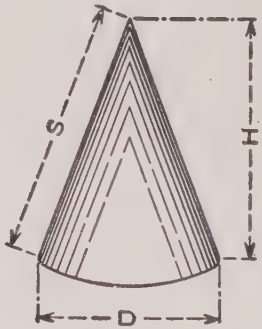


Fig. 102

To find the cubic contents of a cylinder—Fig. 101. Multiply the area of one end, by the length of the cylinder, the product will be the cubic contents of the cylinder.

To find the superficial area of a cylinder. Multiply the circumference of one end by the length of the cylinder and add to the product the area of both ends.

Cylinder: Cubic contents=.7854 (D+H)
Superficial area=1.5708D (2H+D)

To find the cubic contents of a cone—Fig. 102. Multiply the square of the base by the perpendicular height and by .2618.

To find the superficial area of a cone. Multiply the circumference of the base by one-half the slant height and add to the product the area of the base.

Cone: Cubic contents=.2618 ($D^2 \times H$)
Superficial area=.7854 (2S+D)

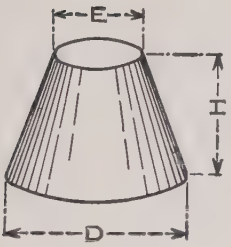


Fig. 103

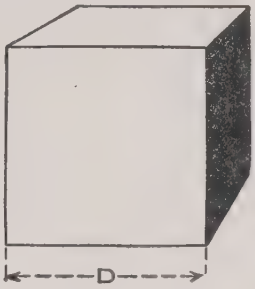


Fig. 104

To find the cubic contents of the frustum of a cone—Fig. 103. To the sum of the areas of the two ends of the frustum, add the square root of the product of the diameters of the two ends, this result multiplied by one-third of the perpendicular height of the frustum will give the cubic contents.

To find the superficial area of the surface of the frustum of a cone. Multiply the sum of the diameters of the ends by 3.1416 and by half the slant height. Add to the result the area of both ends and the sum of the two will be superficial area.

Frustum of cone:

$$\text{Cubic Contents} = \frac{H(.2618\ (E^2+D^2)\sqrt{E\times D})}{3}$$

$$\text{Superficial area} = 3.1416S\left(\frac{D+E}{2}\right)+.7854(E^2+D^2)$$

$$S = \sqrt{\left(\frac{D-E}{2}\right)^2+H^2}$$

To find the contents of a cube—Fig. 104. The contents of a cube equal the cube of its diameter.

To find the superficial area of a cube. The superficial area of a cube is equal to six times the square of its diameter.

Cube: Cubic contents$=D^3$
Superficial area$=6D^2$

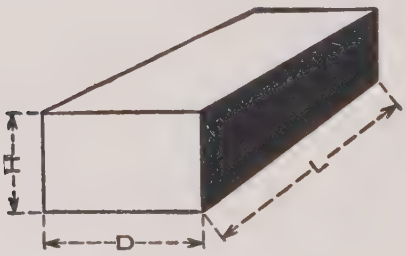


Fig. 105

To find the cubic contents of a rectangular solid—Fig. 105. Multiplying together the length, width and height will give the cubic contents of the rectangular solid.

To find the superficial area of a rectangular solid. Multiply the width by the sum of the height and length and add to it the product of the height multiplied by the

length, twice this sum is the superficial area of the rectangular solid.

Rectangular solid:

Cubic contents=D×H×L

Superficial area=2 (D (H+L) +HL)

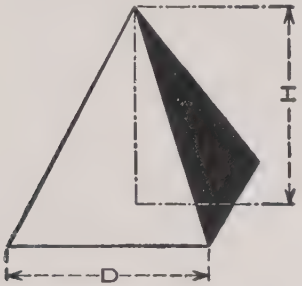


Fig. 106

To find the cubic contents of a pyramid—Fig. 106. Multiply the area of the base by one-third the perpendicular height and the product will be the cubic contents of the pyramid.

To find the superficial area of a pyramid. Multiply the circumference of the base by half the slant height and to this add the area of the base, the sum will be the superficial area.

$$\text{Pyramid: Cubic Contents} = \frac{D^2 \times H}{3}$$

$$\text{Superficial area} = \left(\frac{D4+S}{2} + 4D\right)$$

$$S = \sqrt{\frac{D^2}{4} + H^2}$$

MENSURATION OF TRIANGLES.

To find the base of a right-angle triangle when the perpendicular and the hypothenuse are given—Fig. 107.

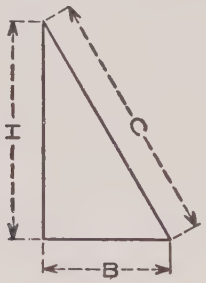


Fig. 107

Subtract the square of the perpendicular from the square of the hypothenuse, the square root of the difference is equal to the length of the base.

$$\text{Base} = \sqrt{\text{Hypothenuse}^2 - \text{Perpendicular}^2} \text{ or } B = \sqrt{C^2 - H^2}$$

To find the perpendicular of a right-angle triangle when the base and hypothenuse are given. Subtract the square of the base from the square of the hypothenuse, the square root of the difference is equal to the length of the perpendicular.

$$\text{Perpendicular} = \sqrt{\text{Hypothenuse}^2 - \text{Base}^2} \text{ or } H = \sqrt{C^2 - B^2}$$

To find the hypothenuse of a right-angle triangle when the base and the perpendicular are given. The square root of the sum of the squares of the base and the perpendicular is equal to the length of the hypothenuse.

$$\text{Hypothenuse} = \sqrt{\text{Base}^2 + \text{Perpendicular}^2}$$

$$C = \sqrt{B^2 + H^2}$$

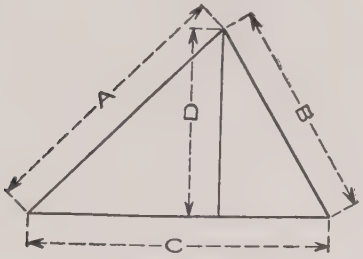


Fig. 108

To find the perpendicular height of any oblique angled triangle—Fig. 108. From half the sum of the three sides of the triangle, subtract each side severally. Multiply the half sum and the three remainders together and twice the square root of the result divided by the base of the triangle will be the height of the perpendicular.

$$D = \frac{2\sqrt{S(S-A)\ (S-B)\ (S-C)}}{C}$$

$$S = \frac{\text{Sum of sides}}{2}$$

To find the area of any oblique angled triangle when only three sides are given. From half the sum of the three sides, subtract each side severally. Multiply the half sum and the three remainders together and the square root of the products is equal to the area required.

$$\text{Area} = \sqrt{S(S-A)\ (S-B)\ (S-C)}$$

To find the height of the perpendicular and the two sides of any triangle inscribed in a semi-circle, when the base of the triangle and the location of the perpendicular are given—Fig. 109.

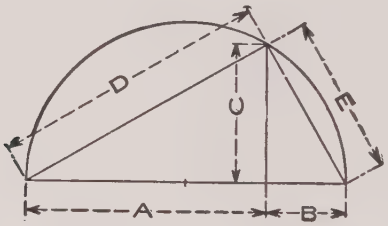


Fig. 109

$$A=\frac{C^2}{B} \qquad B=\frac{C^2}{A} \qquad C=\sqrt{A\times B}$$

$$D=\sqrt{A(A+B)} \qquad E=\sqrt{B(A+B)}$$

THE MECHANICAL POWERS

Mechanical Powers consist of simple mechanical devices whereby weights may be raised or resistances overcome with the exertion of less power than would be necessary without them.

They are six in number: The *lever,* the *wheel* and *pinion,* the *pulley,* the *inclined plane,* the *wedge,* and the *screw.* Properly two of these comprise the whole, namely, the *lever* and the *inclined plane,*—the wheel and pinion being only a lever of the first kind, and the pulley a lever of the second, the wedge and screw being also similarly allied to that of the inclined plane. Although such seems to be the case, yet they each require, on account of their various modifications, a different rule of calculation adapted expressly to the different circumstances in which they are required to act.

The primary elements of machinery are therefore two only in number, the *lever and the inclined plane.*

THE LEVER.

Levers, according to the method of application, are of the first, second, or third kind. Although levers of equal lengths produce different effects, the general principles of estimation in all are the same, namely, the power is to the weight, as the distance of one end of the fulcrum is to the distance of the other end to the same point.

In a lever of the first kind the fulcrum is between the power and the weight, as in Fig. 110. A pair of pliers or scissors are double levers of the first kind.

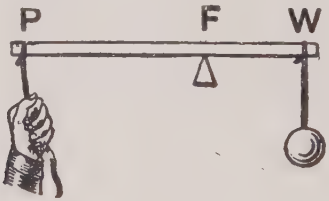


Fig. 110

In a lever of the second kind, the weight is between the power and the fulcrum, as in Fig. 111. A wheel-barrow, or the oars of a boat where the water is considered the fulcrum, and a door, represent levers of the second kind.

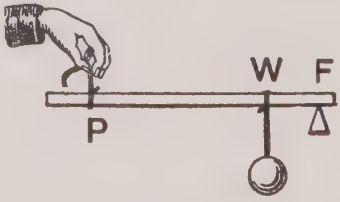


Fig. 111

In a lever of the third kind, the power is between the fulcrum and the weight, as in Fig. 112. Levers of the third kind are instruments as tongs, shears, etc.

In the *first kind,* the power is to the weight, as the distance W F is to the distance F P.

In the *second,* the power is to the weight, as the distance F W is to that of F P; and,

In the *third,* the weight is to the power, as the distance F P is to that of F W.

To find the power. Multiply the weight by its distance from the fulcrum, and divide by the distance of the weight from the fulcrum.

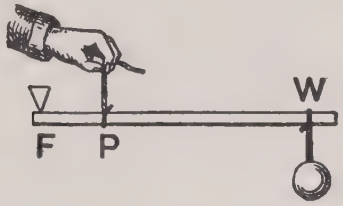


Fig. 112

To find the weight. Multiply the power by its distance from the fulcrum, and divide by the distance of the weight from the fulcrum.

To find the distance of the power from the fulcrum. Multiply the weight by its distance from the fulcrum, and divide by the power.

To find the distance of the weight from the fulcrum. Multiply the power by its distance from the fulcrum, and divide by the weight.

Let P be the power, F the fulcrum and W the weight, then for a lever of the first kind. (Fig. 110.)

$$P = W\frac{FW}{FP} \qquad W = P\frac{FP}{FW}$$

And for a lever of the second kind. (Fig. 111.)

$$P = W\frac{FW}{FP} \qquad W = P\frac{FP}{FW}$$

And for a lever of the third kind. (Fig. 112.)

$$P = W\frac{FW}{FP} \qquad W = P\frac{FP}{FW}$$

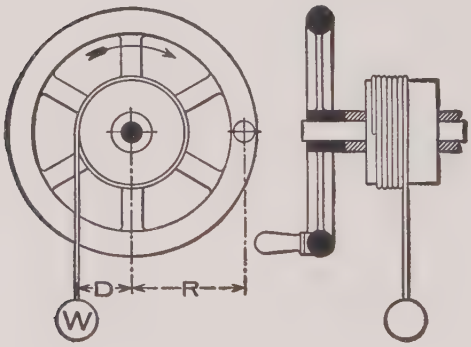


Fig. 113

THE WHEEL AND PINION.

The mechanical advantage of the wheel and pinion system, Fig. 113, is as the velocity of the weight to the velocity of the power, and being only a modification of the first kind of lever, it of course partakes of the same principles.

To find the power. Multiply the weight by the radius of the drum, and divide by the radius of the wheel.

To find the radius of the wheel. Multiply the weight by the radius of the drum, and divide by *the power.*

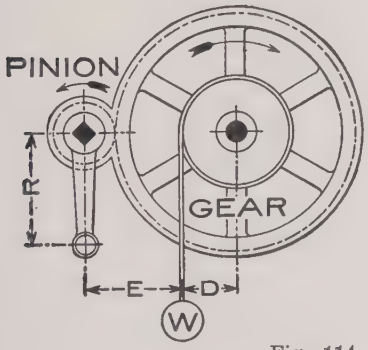
To find the radius of the drum. Multiply the power by the radius of the wheel, and and divide by the weight.

To find the weight. Multiply the power by the radius of the wheel, and divide by the radius of the drum.

Let W be the weight, D the radius of the drum, R the radius of the wheel and P the power required to lift the weight, then for a *Wheel and Drum system*: (Fig. 113.)

$$P=\frac{W\times D}{R} \qquad D=\frac{P\times R}{W} \qquad R=\frac{W\times D}{P} \qquad W=\frac{P\times R}{D}$$

For a *Crank, Pinion and Gear and Drum system*: (Fig. 114.)



$$P=\frac{W\times D\times P}{R\times G} \qquad W=\frac{P\times R\times G}{D\times P}$$

Fig. 114

To determine the amount of effective power produced from a given power by means of a crank, pinion and gear, and drum system. Multiply the diameter of the circle described by the crank or turning handle by the number of revolutions of the pinion to one of the wheel. Divide the product by the diameter of the drum and the

quotient is the ratio of the effective power to the exertive force. Fig. 114.

Given any two parts of a crank, pinion and gear, and drum system, to find the third, that shall produce any required proportion of mechanical effect. Multiply the two given parts together, and divide the product by the required proportion of effect, the quotient is the dimensions of the other part.

$$D=\frac{P\times R\times G}{W\times P} \qquad R=\frac{W\times D\times P}{P\times G}$$

P=Either pitch diameter or number of teeth in the pinion.

G=Either pitch diameter or number of teeth in the gear.

Let E be the ratio of the *effective power* to the *effective force* produced, then

$$E=3.1416\frac{R\times G}{D\times P}.$$

THE PULLEY OR SHEAVE.

The pulley or sheave is a wheel over which a rope is passed to transmit the force applied to the cord in another direction. There are two kinds of pulleys, the one turning on fixed centers, the other turning on traversing centers.

The fixed or stationary pulley (Fig. 115). This acts like a lever of the first kind. It affords no mechanical advantage, and merely changes the direction of the force, and does not alter its intensity, but it affords

great facilities in the application of force, as it is easier to pull downwards than upwards. In this class of pulley the power is equal to the weight to be raised.

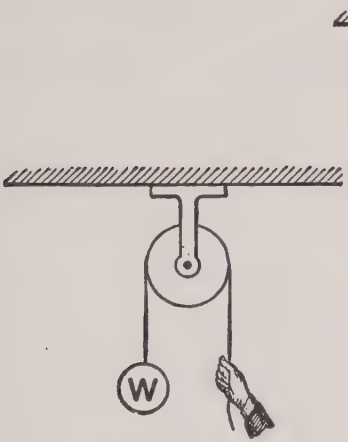


Fig. 115

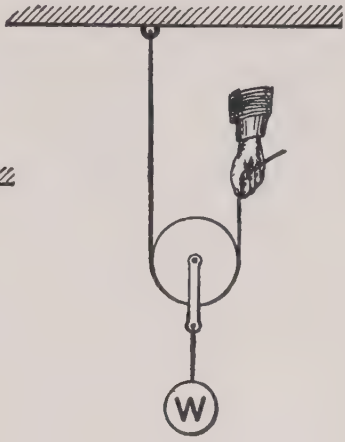


Fig. 116

The movable pulley (Fig. 116). This acts like a lever of the second kind. One end of the rope is suspended to a fixed point, as a fulcrum, in a beam, and the weight is attached to the axis of the pulley. This kind of pulley doubles the power at the expense of the speed, and the product of the power by the diameter of the pulley, is equal to the product of the weight by the radius of the pulley.

A movable pulley acting as a lever of the third kind is shown at Fig. 117. One end of the cord is fixed to a floor, and the weight is attached to the other end, the power being applied to the axis. The power is equal to twice the weight, and the product of the power by

the radius of the pulley is equal to the product of the weight by the diameter of pulley. In the arrangement shown at Fig. 118 the power is equal to one-half the weight.

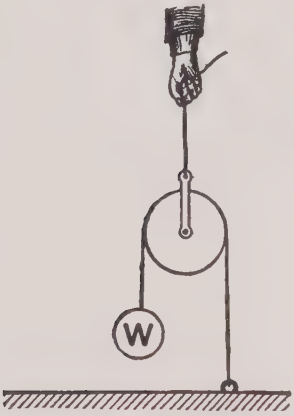


Fig. 117

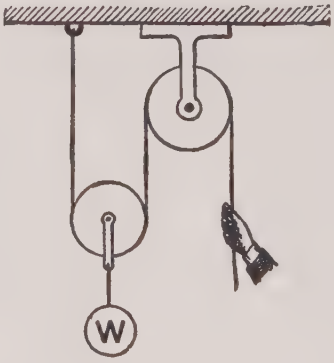


Fig. 118

A combination of movable pulleys with separate and parallel cords is shown at Fig. 119. Each system reduces the resistance to the extent of one-half, hence the power may be found by dividing and subdividing the weight successively by 2, as many times as there are movable pulleys. The weight may be found by multiplying the power successively by 2, as many times as these are movable pulleys.

To find the power. Divide the weight to be raised by the number of cords leading to, from, or attached, to the power block. The quotient is the power required to produce an equilibrium, provided friction did not exist. When the fixed end of the rope is attached to the fixed

block, the power may be found by dividing the weight by twice the number of movable pulleys. When the fixed end of the rope is attached to the movable block, the power may be found by dividing the weight by twice the number of movable pulleys plus 1.

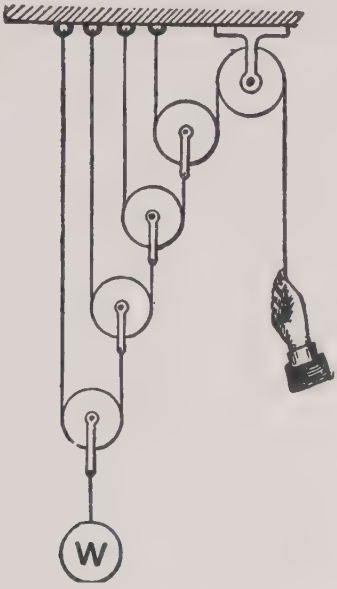


Fig. 119

To find the number of sheaves or pulleys required. Divide the power to be raised by the power to be applied; the quotient is the number of sheaves in, or cords attached to the rising block.

To find the weight that will be balanced by a given power. When the rope is attached to the fixed block,

multiply the power by twice the number of movable pulleys.

When the rope is attached to the movable block multiply the power by twice the number of movable pulleys plus 1.

THE INCLINED PLANE.

The inclined plane (Fig. 120) is properly the second elementary power, and may be defined the lifting of a load by regular instalments. In principle it consists of

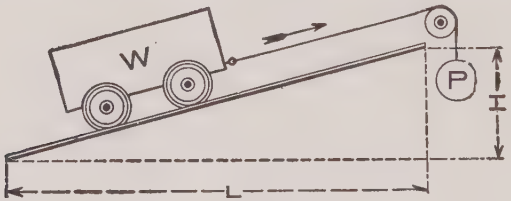


Fig. 120

any right line not coinciding with, but laying in a sloping direction to, that of the horizon, the standard of comparison of which commonly consists in referring the rise to so many parts in a certain length or distance, as 1 in 100, 1 in 200, etc., the first number representing the perpendicular height, and the latter the horizontal length in attaining such height, both numbers being of the same denomination, unless otherwise expressed.

In using an inclined plane for the purpose of raising loads to a higher level, the power is applied parallel to the inclined plane, and the weight is raised in opposition to gravity, the work done on it is expressed by the product of the weight and the vertical height of the inclined plane.

The advantage gained by the inclined plane, when the power acts in a parallel direction to the plane, is as the length to the height.

To find the power. Multiply the weight by the height of the plane, and divide by the slant length. The quotient is the power.

To find the weight. Multiply the power by the slant length of the plane, and divide by the height.

To find the height of the inclined plane. Multiply the power by the slant length, and divide by the weight.

To find the slant length of the inclined plane. Multiply the weight by the height of the plane, and divide by the power.

Let W be the weight to be drawn up the inclined plane, H the height and S the slant length of the incline. If P be the power required to draw the weight W up the inclined plane, then

$$P=\frac{W\times H}{S} \qquad W=\frac{P\times S}{H} \qquad H=\frac{P\times S}{W}$$

$$S=\sqrt{L^2+H^2}$$

THE WEDGE.

The wedge is a double inclined plane, consequently its principles are the same. When two bodies are forced asunder by means of the wedge in a direction parallel to its head: Multiply the resisting power by half the thickness of the head or back of the wedge, and divide the product by the length of one of its slant sides. The quotient is the force required equal to the resistance.

F=Force required. P=Resisting power.

$$F = \frac{P \times H}{25} \qquad P = \frac{F \times 25}{H} \quad \text{(Fig. 121.)}$$

$$S = \text{Slant side of wedge} = \sqrt{\frac{H^2}{4} + L^2}$$

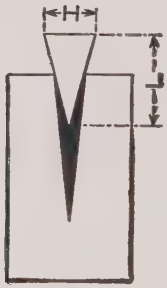


Fig. 121

When only one of the bodies is movable, the whole breadth of the wedge is taken for the multiplier, and the following rules are for such wedges, acting under pressure only on the head of the wedge, or at the point of the wedge by drawing.

To find the transverse resistance to the wedge or weight. Multiply the power by the length of the slant side of the wedge, and divide by the breadth of the head.

To find the power. Multiply the weight or transverse resistance by the breadth of the head and divide by the length of the slant side of the wedge.

To find the length of the slant side of the wedge. Multiply the weight by the breadth of the wedge and divide by the power.

To find the breadth of the wedge. Multiply the power by the length of the slant side of the wedge, and divide by the weight.

F=Force required. P=Resisting power.

$$F=\frac{P\times H}{S} \qquad P=\frac{F\times S}{H}$$ (Fig. 122.)

S=Slant side of wedge=$\sqrt{H^2+L^2}$

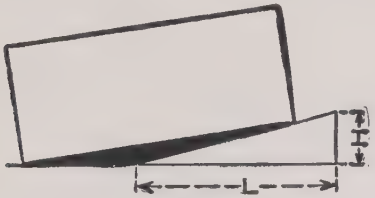


Fig. 122

Note.—For all practical purposes the length L may be used instead of the slant S of the side.

THE SCREW.

The screw, Fig. 123, in principle, is that of an inclined plane wound around a cylinder which generates a spiral of uniform inclination, each revolution producing a rise or traverse motion equal to the pitch of the screw, or distance between two consecutive threads. The pitch being the height or angle of inclination, and the circumference the length of the plane when a lever is not applied. The lever being a necessary qualification of the screw, the circle which it describes is taken, instead of the screw's circumference, as the length of the plane,

the mechanical advantage is therefore as the circumference of the circle described by the lever where the power acts, is to the pitch of the screw, so is the force to the resistance.

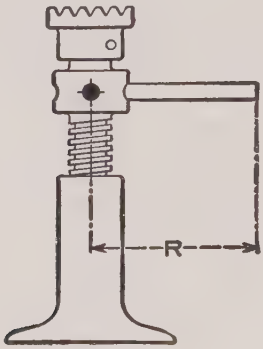


Fig. 123

As the circumference of a circle is equal to the radius multiplied by twice 3.1416, or 6.2832, hence the following rules for the screw.

To find the power. Multiply the weight by the pitch of the screw and divide by the product of the radius of the handle by 6.2832.

To find the weight. Multiply the power by the product of the radius of the handle by 6.2832 and divide by the pitch of the screw.

To find the pitch of the screw. Multiply the power by the product of the radius of the handle by 6.2832 and divide by the weight.

To find the length or radius of the handle. Multiply the weight by the pitch of the screw and divide by the product of the power by 6.2832.

P=Lifting power of jack. R=Length of lever.
F=Force required at end of lever.
N=Number of threads per inch of jack screw.

$$P = 6.283(N \times R \times F) \qquad F = \frac{P}{6.283(N \times R)}$$

$$N = \frac{P}{6.283(R \times F)} \qquad R = \frac{P}{6.283(N \times F)}$$

THE DEVELOPMENT OF CURVES

The Spiral is a curve generated by a point moving in a plane about a center from which its distance is continually increasing.

Imagine a right line, AB, Fig. 124, free to revolve in the plane of the paper about one of its extremities, A, as an axis. Also, conceive a point free to move on this

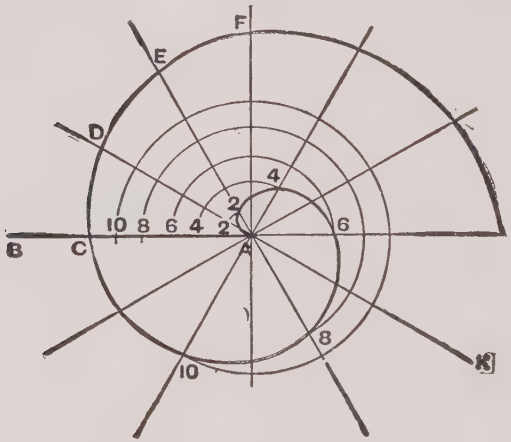


Fig. 124—The Spiral of Archimedes

line. Three classes of lines may now be derived in the following manner: If the line be stationary and the point moves, a straight line will be generated. If the point be stationary and the line revolves, a circle will

be generated. If both point and line move, a spiral will be generated. By varying the relative motion of point and line the character of the curve will be changed and the various classes of spirals described.

The Spiral of Archimedes—Fig. 124. If the motion of the line and the point be uniform the equable spiral will be generated. The line AB is called the radius vector, and the radial distance traversed by the point during one revolution of the radius vector is called the

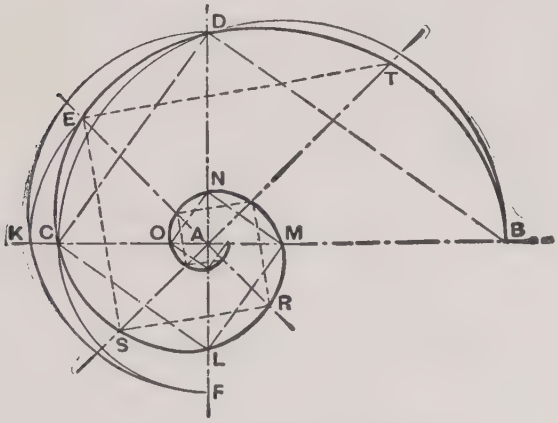


Fig. 125—The Logarithmic Spiral

pitch. Twelve successive positions of the radius vector are shown by AC, AD, AE, etc. The distance of the point from the center being increased by one-twelfth of the pitch, AC, for each one-twelfth of a revolution of the radius vector. In practice, determine at least twenty-four points, and lightly sketch the curve freehand.

The Logarithmic Spiral. The construction of this curve is based on the principle that any radius vector, as

AD, Fig. 125, which bisects the angle between two other radii, as AC and AB, is a mean proportional between them, that is, $(AD)^2 = AC \times AB$. This spiral is called equiangular because the angle between any radius vector and the tangent to the curve at its extremity is constant.

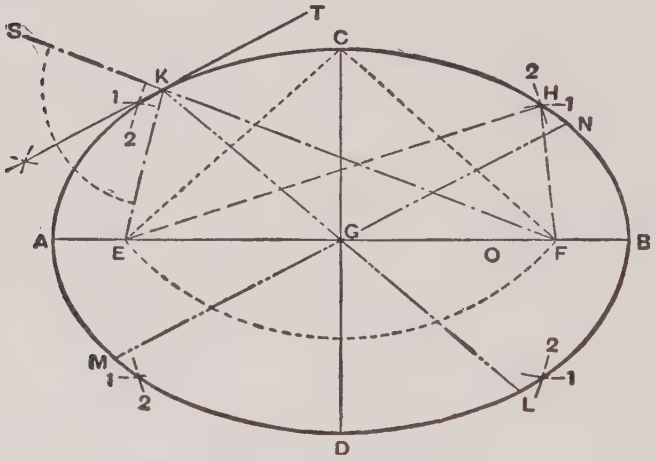


Fig. 126—The Ellipse—1

If B and C are points in the spiral and the ratio of AC and AB be given, the intermediate point D may be obtained by describing a semicircle on BC as a diameter and erecting a perpendicular at A. Its intersection with the semicircle at D will determine the required point, and AD will be a mean proportional between AC and AB. Other points on the curve lying on the diameters BC and DF may be obtained by intersecting these diameters with the perpendiculars CL, LM, MN, etc. Again, having points C and D, bisect DAC and deter-

mine a mean proportional between AC and AD. This may be done by laying off AF equal to AC and determining the mean proportional AK as before. Then lay off AE equal to AK. Other points on the diameters ER and T may be obtained by perpendiculars as indicated.

The Ellipse is a curve generated by a point moving in a plane so that the sum of the distances from this point to two fixed points shall be constant. Fig. 126, if EKF be a cord fastened at its extremities, E and F, and held taut by a pencilpoint at K, it may be seen that as motion is given to the point it will be constrained to move in a fixed path dependent on the length of the cord. When the pencil-point is at B, one segment of the cord will equal BE and the other BF, their sum being the same as KE plus KF, and also equal to AB. The fixed points E and F are called the foci. They lie on the longest line that can be drawn terminating in the curve of the ellipse. The line is known as the major axis, and the perpendicular to it at its middle point, also terminating in the ellipse, is the minor axis. Their intersection is called the center of the ellipse, and lines drawn through this point and terminating in the ellipse are known as diameters. When two such diameters are so related that a tangent to the ellipse at the extremity of one is parallel to the second, they are called conjugate diameters. KL and MN are two such diameters.

In order to construct an ellipse it is generally necessary that either of the following be given: The major and minor axes, either axis and the foci, two conjugate diameters, a chord and axis.

A series of points must be chosen that the sum of the

distances from either of them to the foci must equal the major axis. Thus, HE+HF must equal CE+CF, or KF+KE, each being equal to AB. If the major axis and the foci be given to draw the curve, points may be determined as follows: From E, with any radius greater than AE and less than EB, describe an arc.

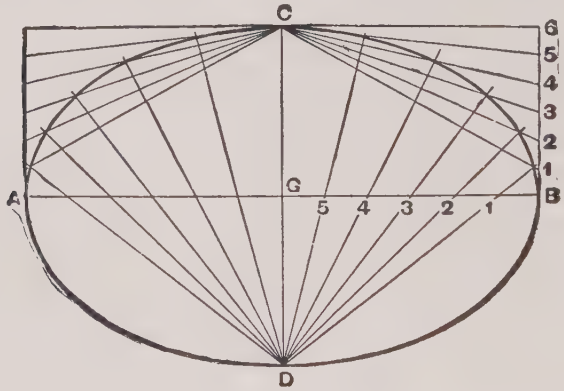


Fig. 127—The Ellipse—2

From F, with a radius equal to the difference between the major axis and the first radius, describe a second arc cutting the first. The points of intersection of these arcs will be points, the sum of whose distances from the foci will equal the major axis, and therefore points of an ellipse. Similarly find as many points as may be necessary to enable the curve to be drawn free-hand.

Having given the major and minor axes, find the foci by describing, from C as a center, an arc with a radius equal to one-half the major axis. The points of intersection with the major axis will be the foci, and this must be so since the sum of these distances is equal to

the major axis, and the point C being midway between A and B the two lines CE and CF must be equal. Again, if the major axis and foci are given, with a radius equal to one-half this axis describe arcs from the foci cutting the perpendicular drawn at the midde point of the major axis and thus obtain the minor axis. Having the two axes proceed as before.

A tangent to an ellipse may be drawn at any point, K, by producing FK and dissecting the angle SKE, the bisecting line, KT, will be the required tangent.

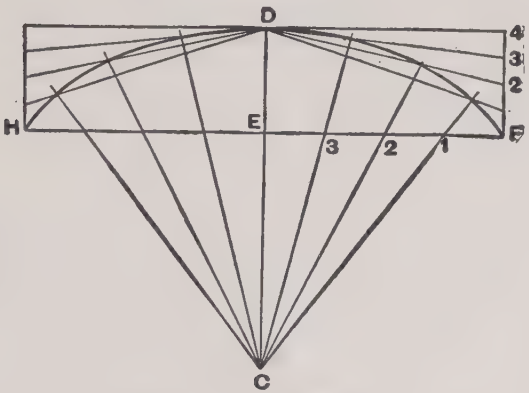


Fig. 128—The Ellipse—3

Having given the major and minor axes in Fig. 127. From the extremity of the major axis, draw B6 parallel and equal to the minor axis, and divide it into any number of equal parts, in this case six. Divide BG into the same number of equal parts. Through points 1, 2, 3, etc., on B6, draw lines to extremity C of the minor axis. From D, the other extremity of the minor axis, draw

lines through points 1, 2, 3, etc., on BG, intersecting the above lines in points which will lie in the required ellipse. Construct the remainder of the ellipse in the same manner.

Having given an axis CD and chord FH as in Fig. 128. From F draw F4 parallel to CD, divide it into any number of equal parts, in this case four. Divide the half chord FE into the same number of equal parts, through these points and extremities of given axis draw intersecting lines as before, thereby obtaining the elliptical arc FD. Construct opposite side in the same manner.

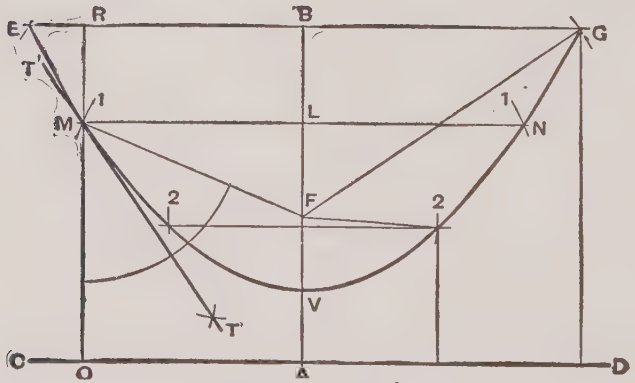


Fig. 129—The Parabola

The *Parabola* is a curve generated by a point moving in a plane so that its distance from a fixed point shall be constantly equal to its distance from a given straight line. Point F in Fig. 129 is the focus, CD is the given straight line called the directrix, and AB, a perpendicu-

lar to CD through F, is the Axis V, the intersection of the axis with the curve, is the vertex, and it must be equidistant from the focus and the directrix.

Having given the focus F and the directrix CD. Bisect FA to find the vertex V. Through any point on the axis, as L, draw MN parallel to the directrix and with radius LA describe arc 1 from focus F as center, intersecting line MN at points M and N. These are points in the parabola. Similarly obtain other points and draw the required curve. A tangent to the curve may be drawn at any point M by drawing MO parallel to the axis and bisecting the angle OMF. MT is the required tangent at the point M.

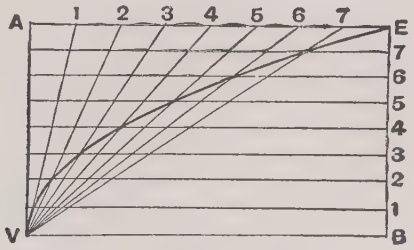


Fig. 130

Since the angle T′MR is equal to the angle TMF, it follows that MR would be the direction of a ray of light emitted from the focus F and reflected from the parabola at M.

Having given the abscissa VB, and the ordinate BE in Fig. 130. Draw AE parallel and equal to VB. Divide AE and BE into the same number of equal parts.

From the divisions on BE draw parallels to the axis and from the divisions on AE draw lines converging to the vertex V. The intersection of these lines, 1 and 1, 2 and 2, etc., will determine points in the required curve. In like manner obtain the opposite side.

MACHINE DESIGN

Box Couplings. The coupling illustrated in Fig. 185 consists of a solid box made of cast-iron, bored out to fit the shafts, whose ends are made to butt together inside the box. The box may be secured to the shafts by means of a sunk key, which extends the whole length of the box. A better arrangement is to use two keys, both driven from the same end of the box, as shown in Fig. 185. With two keys it is not so important that the keyways in the shafts have exactly the same depth. Two keys may be more tightly driven in, and are easier to

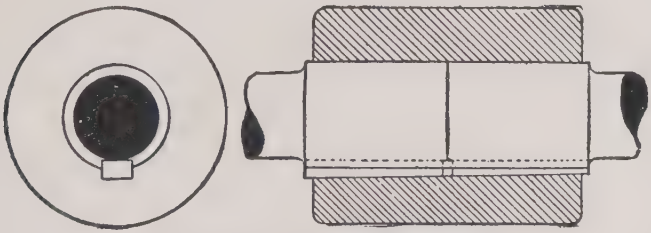


Fig. 185—Box Shaft Coupling

drive back than a single key of the same total length. There must, however, be a clearance space between the head of the forward key, and the point of the hind one, so as to ensure that the latter is driven in with the same degree of tightness as the former. In driving the keys back also, they can be started separately, and therefore

more easily, when this clearance space is allowed. Fig. 185 shows the ends of the shafts enlarged where the keyways are cut, so that the latter do not weaken the shafts, the amount of the enlargement being such that the bottom of the keyway touches the outside of the shaft.

Split Couplings. This form of coupling, which is shown in Fig. 186, is very easily put on or taken off, it has no projecting parts, the bolts being completely covered, and, like the solid couplings, it may be used as a

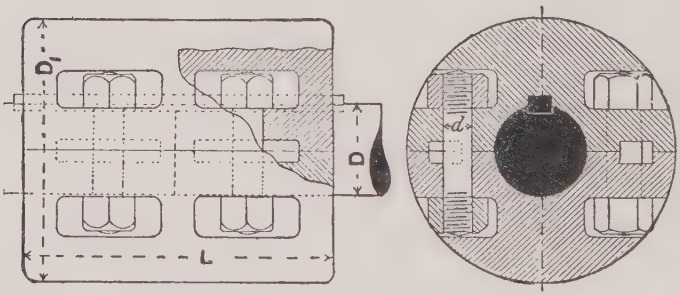


Fig. 186—Split Shaft Coupling

driving pulley, or a driving pulley may be placed on it. In making this coupling the faces for the joint between the two halves of the box are first planed. The bolt-holes are then drilled and the two halves bolted together with pieces of paper between them; then the muff is bored out to the exact size of the shaft. When the paper is removed and the box put on the shaft and bolted up, the box gripes the shaft firmly. The key has no taper and should fit on the sides only.

DIMENSIONS OF SPLIT COUPLINGS.

Diam. of shaft, D	1½	1¾	2	2¼	2½	2¾	3	3½	4
Diam. of Box, D1	4½	5¼	5¾	6	6½	7¼	7⅞	9¾	10½
Length of Box, L	6	7	8	9	10	11	12	14	16
Diam. of Bolts, *d*	½	⅝	⅝	⅝	¾	⅞	⅞	⅞	⅞
Number of Bolts	4	4	4	4	4	4	4	6	6

Cast-Iron Flange Coupling. In this form of coupling, which is shown in Fig. 187, there are cast-iron pieces or bosses, provided with flanges which are keyed to the

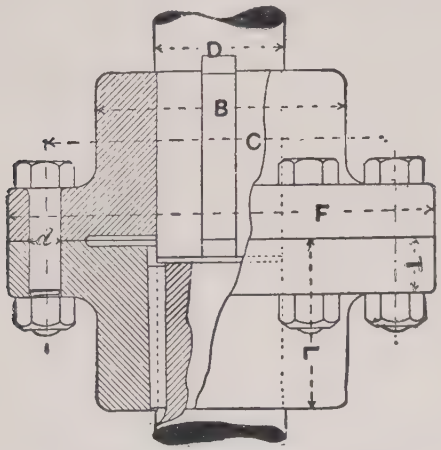


Fig. 187—Cast-iron Flange Coupling

ends of the shafts to be connected. These flanges are fastened together by means of bolts and nuts as shown. Sometimes the shaft is enlarged where it enters the coupling, so as to allow for the weakening effect of the

keyway, but more frequently it is parallel throughout, or very slightly reduced, as shown in Fig. 187, so as to form a shoulder, which prevents the shaft going farther into the coupling.

DIMENSIONS OF CAST-IRON FLANGE COUPLINGS.

Diam. of Shaft D.	Diam. of Flange F.	Thickness of Flange T.	Diam. of Boss B.	Depth at Boss L.	Number of Bolts	Diam. of Bolts d.	Diameter of Bolt Circle C.
1	5 5-8	3-4	2 5-8	2 1-16	3	1-2	4 1-4
1 1-4	6 13-16	13-16	3 1-16	2 5-16	3	5-8	5 1-16
1 1-2	7 1-4	7-8	3 1-2	2 5-8	3	5-8	5 1-2
1 3-4	7 11-16	1	3 15-16	2 7-8	4	5-8	5 15-16
2	8 7-8	1 1-16	4 3-8	3 3-16	4	3-4	6 3-4
2 1-4	9 1-16	1 1-8	4 3-4	3 7-16	4	3-4	7 1-8
2 1-2	10 9-16	1 1-4	5 5-16	3 3-4	4	7-8	8 1-8
2 3-4	11	1 5-16	5 3-4	4 1-16	4	7-8	8 9-16
3	12 3-8	1 7-16	6 1-4	4 5-16	4	1	9 1-2
3 1-4	12 5-8	1 1-2	6 5-8	4 5-8	4	1	9 13-16
3 1-2	13 1-8	1 5-8	7 1-8	4 7-8	4	1	10 5-16
3 3-4	13 9-16	1 11-16	7 9-16	5 3-16	4	1	10 3-4
4	14	1 3-4	8	5 7-16	6	1	11 1-4
4 1-4	14 7-16	1 7-8	8 7-16	5 3-4	6	1	11 5-8
4 1-2	15 5-8	2	8 7-8	6	6	1 1-8	12 1-2
4 3-4	16 1-8	2 1-16	9 3-8	6 5-16	6	1 1-8	13
5	17 5-16	2 1-8	9 13-16	6 5-8	6	1 1-4	13 13-16
5 1-4	17 3-4	2 1-4	10 1-4	6 7-8	6	1 1-4	14 1-4
5 1-2	18 3-16	2 5-16	10 3-4	7 1-4	6	1 1-4	14 11-16
5 3-4	19 1-2	2 7-16	11 1-4	7 7-16	6	1 3-8	15 5-8
6	19 7-8	2 1-2	11 5-8	7 3-4	6	1 3-8	16

Gib-Heads on Keys. When the point of a key cannot be conveniently reached for the purpose of driving it out, the head should be formed as shown in Fig. 188. This is known as a gib-head.

Sliding or Feather Keys. The function of a sliding or feather key is to secure a piece to a shaft, so far as

to prevent the one from rotating without the other, but at the same time allow of a relative motion in the direction of the axis of the shaft. This form of key has no

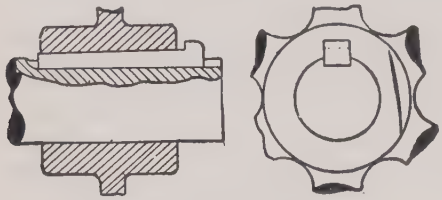


Fig. 188—Gib-head Key

taper, and it is generally secured to the piece carried by the shaft, in which case it is made a sliding fit in the keyway of the shaft. But the feather-key may be fixed to the shaft and made a sliding fit in the piece carried by it.

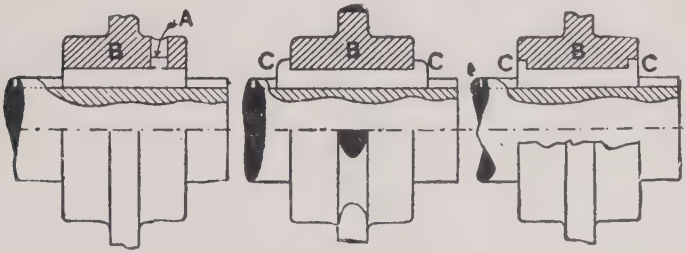


Fig. 189—Sliding Feather Keys

Examples of feather-keys which slide in the key-way of the shaft are shown in Fig. 189. In the left-hand view the key has a projecting pin, A, formed on it, which enters a corresponding hole in the piece B carried by the shaft. In the other two views gib-heads, C, are formed on the ends of the key, which serve the same

purpose as the pin A in the left-hand view, namely, to ensure that the key and the piece B remain or move together.

Fig. 190 shows two examples of feather-keys which are fixed to the shaft. In the left-hand view the key is sunk into the shaft, and in the right-hand view the key is either a saddle-key or a key on a flat, and is secured to the shaft by screws as shown. In these cases the key must be long enough to permit of the necessary sliding motion.

DIMENSIONS OF KEYS

D=diameter of shaft.
B=breadth of key.
T=thickness of sunk key.
T_1=thickness of flat key, also=thickness of saddle-key.
Taper of key ⅛-inch per foot of length.

D.	B.	T.	T_1.	D.	B.	T.	T_1.
¾	5/16	¼	3/16	4	1⅛	½	7/16
1	⅜	¼	3/16	4½	1¼	9/16	½
1¼	7/16	¼	3/16	5	1⅜	⅝	½
1½	½	5/16	3/16	5½	1½	11/16	9/16
1¾	9/16	5/16	¼	6	1⅝	¾	⅝
2	⅝	5/16	¼	7	1⅞	13/16	11/16
2¼	11/16	⅜	¼	8	2⅛	15/16	¾
2½	¾	⅜	5/16	9	2⅜	1	⅞
2¾	13/16	7/16	5/16	10	2⅝	1 1/16	15/16
3	⅞	7/16	5/16	11	2⅞	1 3/16	1 1/16
3½	1	½	⅜	12	3⅛	1¼	1⅛

The proportions of the gib-head for a key, Fig. 188, may be h=thickness of key, l=1½h.

Screwed Cast-Iron Flanges for Wrought-Iron Pipes. In a paper read before the American Society of Mechanical Engineers, Mr. E. F. C. Davis described the joint used for the steam-pipes in the coal mines of the Phila-

delphia and Reading Coal and Iron Company. The pipes are of wrought-iron, and carry cast-iron flanges of the form shown in Fig. 191. The flange is screwed tightly on

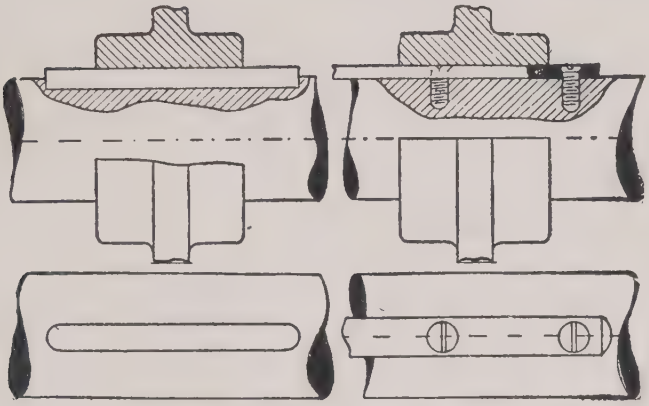


Fig. 190—Fixed Feather-keys

the pipe, and the end of the latter is faced off flush with the facing piece, which is cast on the former. One of the two flanges which are to be bolted together has cast on its face a number of lugs, which have their inner faces bored to fit the facing piece on the other flange, and thus ensure the pipes being in line. The joint is made steam-tight by a gum-joint ring which is placed between the abutting ends of the pipes and within the circle of the lugs, so that the latter keep the ring central.

The following table gives the dimensions in inches adopted for this pipe joint:

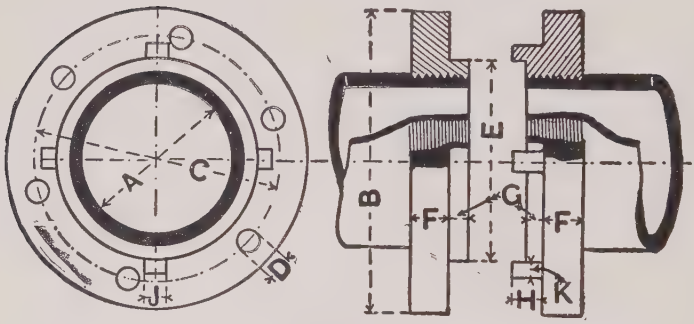


Fig. 191—Cast-iron Flanges for Wrought-iron Pipes

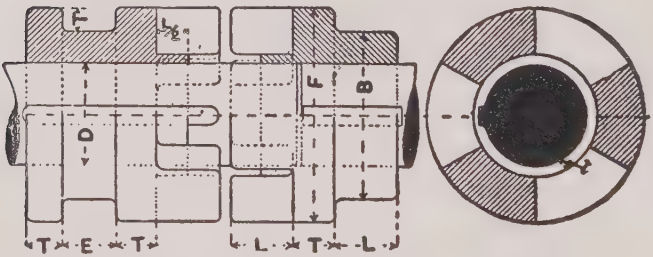


Fig. 192—Square-jaw Detachable Coupling

A.	B.	N_1.	C.	D.	E.	F.	G.	N_2.	H.	J.	K.
3	7¾	4	6	¾	5	¾	½	4	⅞	¾	½
3½	8½	4	6¾	¾	5½	¾	½	4	⅞	¾	½
4	9½	4	7½	⅞	6	⅞	½	4	⅞	¾	½
5	10½	4	8½	⅞	7	⅞	½	4	⅞	¾	½
6	12	6	10	⅞	8	1	½	4	⅞	¾	½
7	13	6	11	⅞	9	1	½	4	⅞	¾	½
8	14	6	12	⅞	10	1¼	½	6	⅞	1	⅝
10	16¼	8	14	1	12	1¼	½	9	⅞	1	⅝

N_1=number of bolts. N_2=number of lugs.

Jaw Coupling. For large and slow moving shafts, the cast-iron jaw coupling shown in Fig. 192 is very simple and effective. In Fig. 192 one-half of the coupling is shown fitted with a feather-key, so that this half may be disconnected from the other or geared with it at pleasure. If the jaws are shaped as shown in Fig. 193, the two halves of the coupling are more easily put into gear, but in this case the motion must always take place in the same direction.

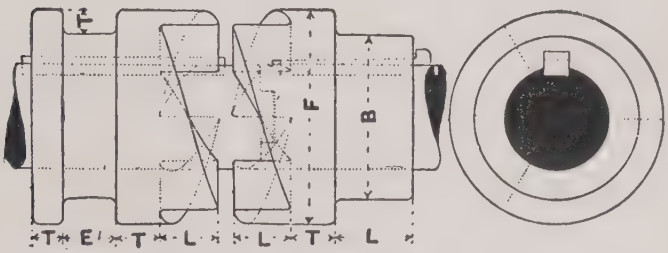


Fig. 193—Spiral-jaw Detachable Coupling

Band Friction Clutch. In the friction clutch, shown in Fig. 194, there is a sleeve, A, which is keyed to the shaft and also attached to the friction band, B, by means of bolts, as shown. The friction band, B, encircles the boss, C, of the wheel or pulley, the latter being loose on the sleeve, A. The band, B, may be bound to, or released from, the boss, C, by the action of right and left-handed screws controlled by the sliding sleeve, D, and levers, E. It is evident that the band, B, moves with the shaft, and that the wheel or pulley will only rotate when the band, B, is tight on the boss, C. When used as a shaft coupling the boss, C, is prolonged and keyed to one of the shafts, the sleeve, A, being keyed to the other.

Cone Friction Clutch. The friction clutch, shown in Fig. 195, although simpler in construction than those which have already been described, is open to the objection that it requires a much greater force to put it into

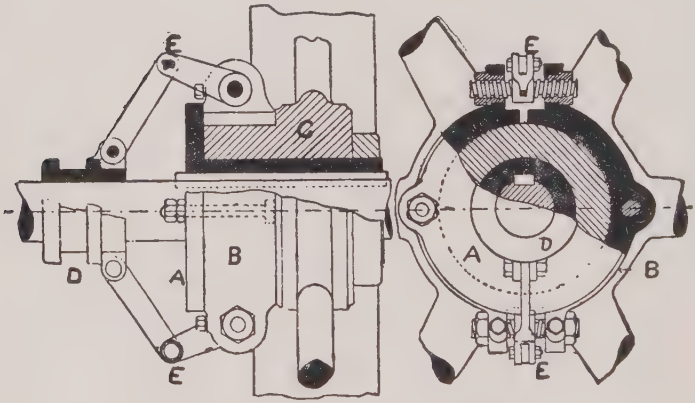


Fig. 194—Band Friction Clutch

gear, and also this force, acting parallel to the axis of the shaft, causes an objectionable end thrust. The mean diameter of the conical part may be from four to eight times the diameter of the shaft, being larger the greater the amount of power which the clutch has to transmit. The inclination a of the slant side of the cone to its axis may vary from four degrees to ten degrees. The other proportions may be as follows:

B=2 D+1.
C=1.5 D.
F=1.8 D.
H=.5 D.
E=.4 D+.4.
T=.3 D+.3.
t=.2 D+.1.

Transmission of Motion by Belts. If motion be transmitted from one pulley to another by means of a thin inextensible belt as in Fig. 196, every part of the latter

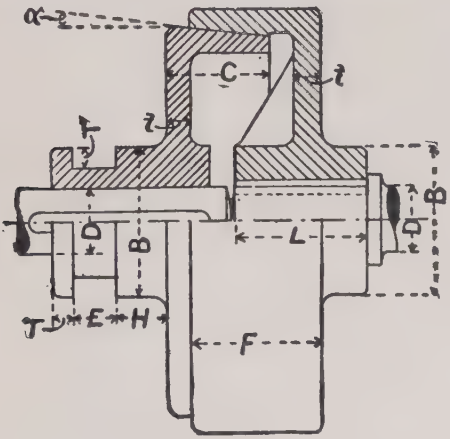


Fig. 195—Cone Friction Clutch

will have the same velocity, and the outer surface of the rim of each pulley will have the same velocity as the belt. This fact enables us to prove the formula con-

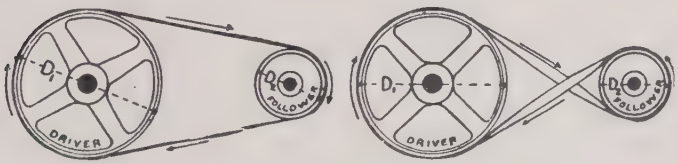


Fig. 196—Belt Transmission of Power

necting the speeds of two pulleys with their diameters. Let D_1 and D_2 be the diameters of the driver and follower respectively, and let N_1 and N_2 denote their speed;

in revolutions per minute. The speed of the outer surface of the rim of the driver$=D_1\times3.1416\times N_1$, and the speed of the outer surface of the rim of the follower$=D_2\times3.1416+N_2$, but each of these is equal to the speed of the belt, therefore, $D_1\times3.1416\times N_1=D_2+3.1416+N_2$; that is, $D_1N_1=D_2N_2$ or $\frac{N_2}{N_1}=\frac{D_1}{D_2}$. This formula is true whether the belt is open or crossed, but the direction of the motion will not be the same in each of these cases, as is shown by the arrows. With an open belt the pulleys rotate in the same direction, while with a cross belt, the pulleys rotate in opposite directions.

Example 1. The driving pulley is 3 feet 6 inches in diameter, and it makes 100 revolutions per minute. The driven pulley is 2 feet 6 inches in diameter. To find its speed.

$$D_1N_1=D_2N_2$$
$$42\times100=30\times N_2$$
$$\text{therefore } N_2=\frac{42\times100}{30}=140 \text{ rev. per minute.}$$

Example 2. A belt moving with a velocity of 1,000 feet per minute passes over a pulley 3 feet 3 inches in diameter. To find the number of revolutions made by the pulley in one minute.

$$\text{Circumferential speed of pulley}=\frac{39\times3.1416\times N}{12}=1000,$$
$$\text{therefore } N=\frac{1000\times12}{39\times3.1416}=97.94 \text{ revolutions per minute.}$$

Split Pulleys. For convenience in fixing pulleys on shafts it is a common practice to make them in halves, which are bolted together as shown in Fig. 197, which

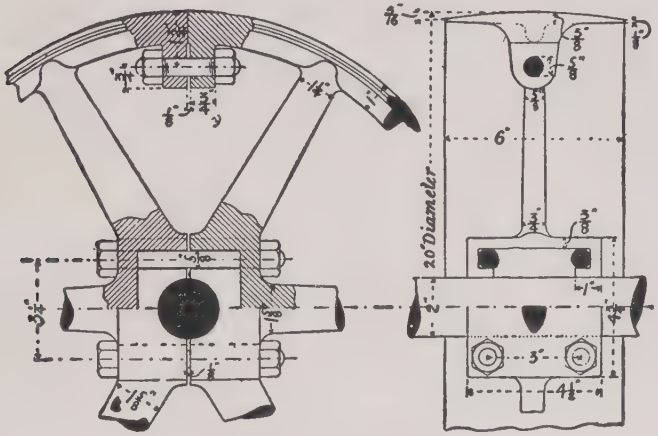


Fig. 197—Split Pulley

illustrates an example of a cast-iron split pulley 20 inches by 6 inches for a 2-inch shaft. By leaving a small clearance space between the two halves of the pulley at the hub, the latter may be tightened on to the shaft so firmly by means of the bolts that no key is necessary.

Wrought-Iron Pulleys. A great number of wrought-iron pulleys are now made, and they possess several important advantages over cast-iron ones. They are much lighter, and, being built up, they are free from the initial strains which exist in cast-iron pulleys due to unequal contraction in cooling. They are also safer at high speeds, not only because wrought-iron has a greater tenacity than cast-iron, but because if a wrought-iron

pulley should burst, it will not fly to pieces like a cast-iron one, since cast-iron is so much more brittle than wrought-iron. Wrought-iron pulleys can be made of any diameter required, but it is when they are of large diameter that their superiority over cast-iron pulleys is most decided.

In a wrought-iron pulley the rim is made of sheet-iron, and riveted to the arms, which are made from bar-iron. The hub is sometimes also made of wrought-iron, but generally it is made of cast-iron.

The differences in the design of wrought-iron pulleys are chiefly in the form of the arms.

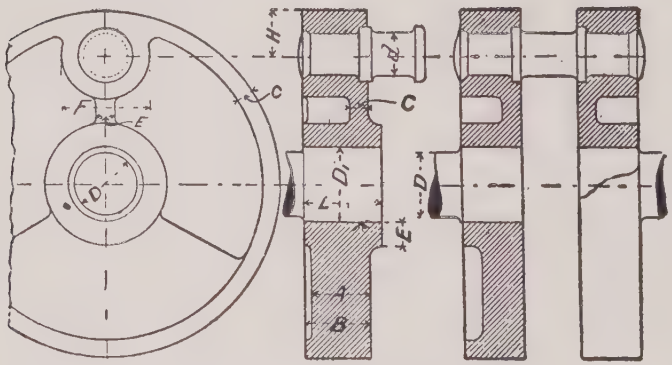


Fig. 198—Crank Disks

Crank-Disks. In small engines, and in engines which run at a high speed, a cast-iron disk frequently takes the place of the ordinary crank. This disk is hollowed out on the side next the crank-pin, as shown in Fig. 198, and the extra weight on the opposite side balances the crank-pin, and the part of the connecting-rod which

revolves with it. Two crank-disks may be used to take the place of a double-armed crank.

The following rules may be used in proportioning crank disks:

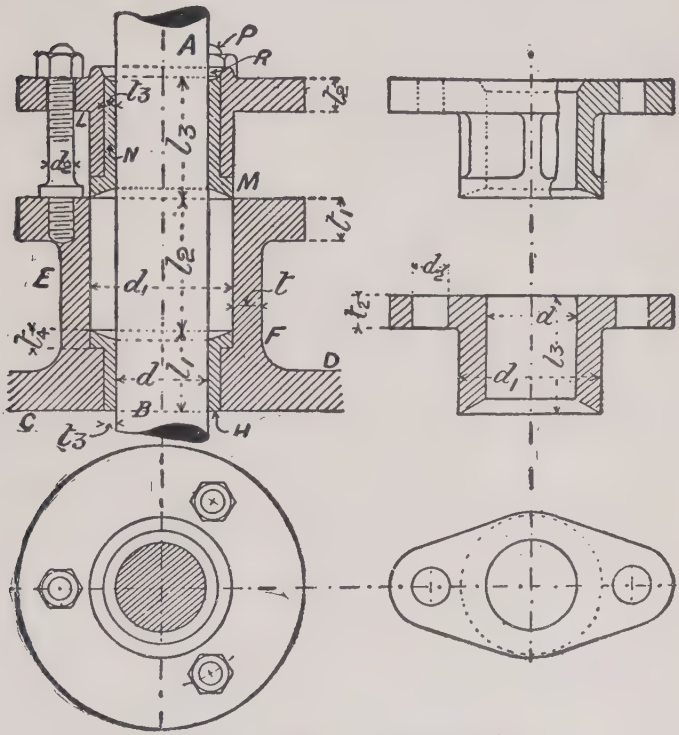


Fig. 199—Piston Rod Stuffing-box

D_1=D to 1.2D.
B=.7D to D.
A=.9B.
C=.25D.
E=.4D
F=2*d*.
H=*d* to 1.5*d*.
L=B to 1.25B.

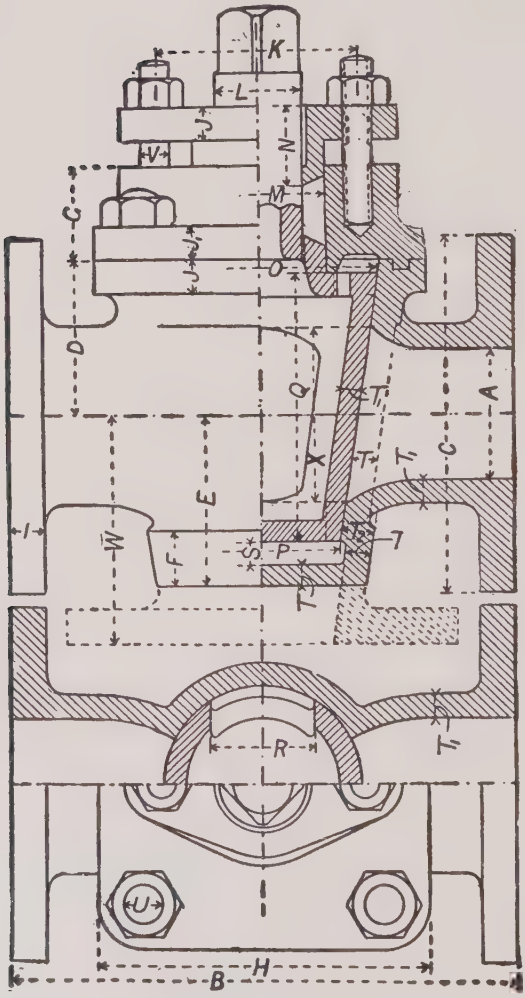


Fig. 200—Steam Cock

A Stuffing-Box is used where a sliding or rotating piece passes through the end or side of a vessel containing a fluid under pressure. The stuffing-box allows the sliding or rotating piece to move freely, without allowing any leakage of the fluid. Applications of the stuffing-box are found in the steam-engine where the piston-rod passes through the cylinder cover, and where the valve spindle passes through the valve casing; also at the trunions of the cylinder in an oscillating engine, and where the shaft of a centrifugal pump passes through the pump casing. Stuffing boxes are also used to permit of the expansion and contraction of steam pipes.

Fig. 199 shows an ordinary form of stuffing-box for the piston-rod of a vertical engine. AB is the piston-rod, CD a portion of the cylinder cover, and EF the stuffing-box. Fitting into the bottom of the stuffing-box is a brass bush, H. The space around the rod AB is filled with packing, of which there are a great many kinds, the simplest being greased hempen rope. The packing is compressed by screwing down the gland LM by means of the bolts P. When more than two bolts are used, the gland flange is generally made circular as shown in the lower left-hand view. For two bolts the gland flange is of the form shown in the lower right-hand view. When the gland is made of cast-iron it is generally lined with a brass bushing. When the gland is of brass no lines are necessary.

Steam Cock—Fig. 200 shows a good example of a cock, suitable for steam or water, and adjacent to it is a table of dimensions, taken from actual practice, for four different sizes. The two parts of the pipe meeting

at the plug may be in the same straight line, or one may proceed from the bottom of the casing, as shown by the dotted lines.

A	1¾	2	2¼	2½	N	1 3/16	1¼	1 5/16	1⅜
B	7¼	7⅞	8½	9	O	3	3⅜	3 13/16	4 3/16
C	5	5¾	6	6⅜	P	2 1/16	2⅜	2 11/16	3
D	1⅞	2 3/16	2⅝	2¾	Q	3½	4	4½	5
E	2 7/16	2¾	2 15/16	3 5/16	R	1 5/16	1½	1 11/16	1 13/16
F	⅞	1	1	1⅛	S	⅜	⅜	⅜	7/16
G	1⅜	1 7/16	1½	1 9/16	T	⅜	⅜	7/16	7/16
H	4½	5¼	5⅝	6	T_1	5/16	5/16	⅜	⅜
I	½	17/32	9/16	⅝	T_2	7/16	7/16	½	½
J	½	17/32	9/16	⅝	U	½	⅝	⅝	⅝
J_1	7/16	15/32	½	9/16	V	½	½	½	⅝
K	2⅞	3¼	3⅜	3 11/16	W	3⅜	3¾	3⅞	4 1/8
L	1¼	1⅜	1 7/16	1½	X	2¼	2 9/16	2⅞	3⅛
M	1⅞	2	2 3/16	2¼					

Shaft Hanger with Adjustable Bearing—Fig. 201 shows an excellent form of hanger and bearing for a shaft. The bearing is very long, and is held at the center of its length on spherical seats, which are formed on the ends of the vertical adjusting screws. The spherical seats permit of a slight angular movement of the bearing, a movement which is necessary with such a long bearing, to ensure that the axis of the bearing coincides with the axis of the shaft. The bearing and the vertical adjusting screws are made of cast-iron.

Locomotive Connecting-Rod End—A modified form of box end is shown in Fig. 202. This is the design adopted on some roads for the large or crank-pin ends of the connecting-rods. The end of the rod is forked, and the open end is closed by a block which is held in position by a bolt. This bolt has a slight taper, and is screwed at both ends and provided with nuts, so that it

may be easily tightened up, and also easily withdrawn. The brasses, which have flanges on each side, are tightened up by a wedge-shaped block attached by two pins

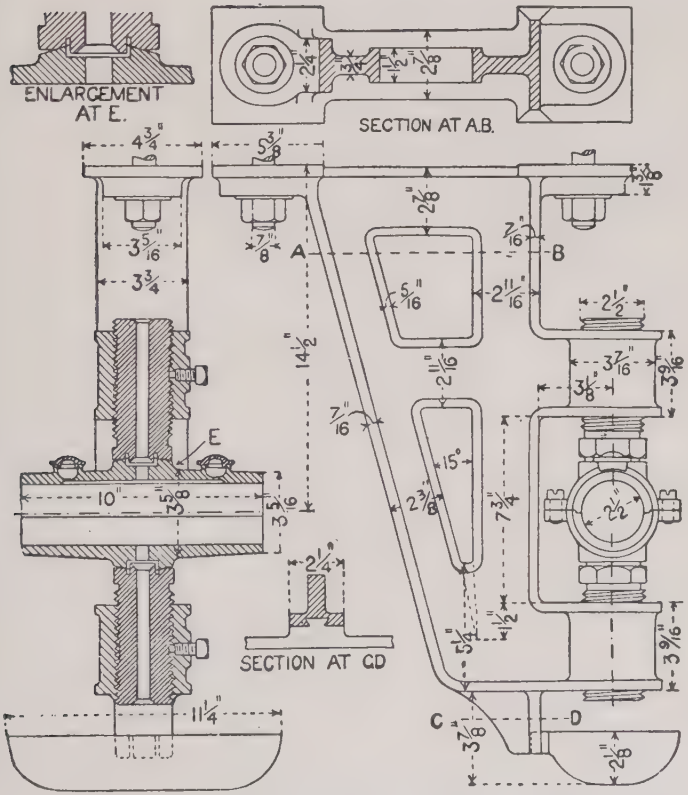


Fig. 201—Shaft-hanger with Adjustable Bearing

to a bolt screwed at both ends.

Since the angular motion of a long connecting-rod

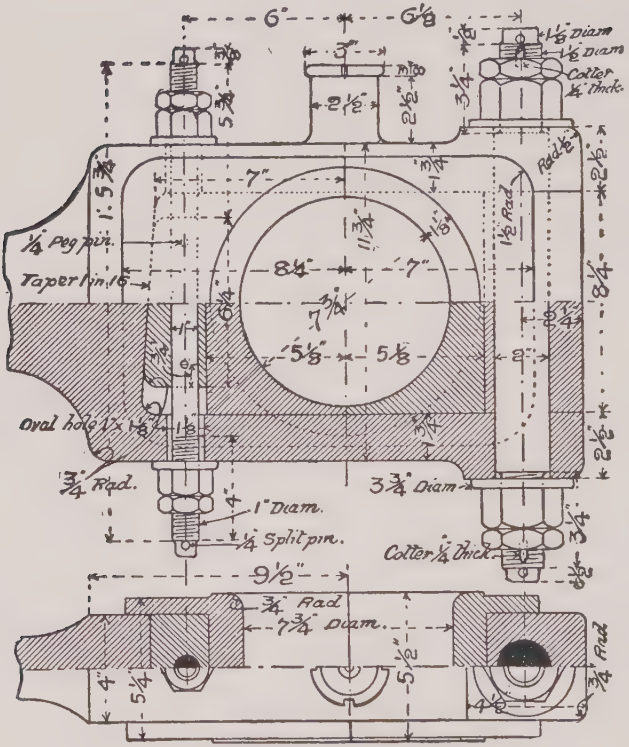


Fig. 202—Locomotive Connecting-rod End

is small, the wear of the brasses at the cross-head end is slight. For this reason the bearing at the cross-head end of locomotive connecting-rods is sometimes made solid.

Locomotive Cross-Head—Fig. 203 shows a form of cross-head used on certain railroads. In this form the cross-head and slide-blocks are made in one piece of cast-

steel. The cross-head pin is tapered, and carries a phosphor bronze bushing or sleeve at its center, upon which works the connecting-rod end, the latter being solid and fitted with a hardened steel bushing. The phosphor bronze bushing is prevented from rotating on the cross-head pin by a sunk key, which also prevents the pin from rotating.

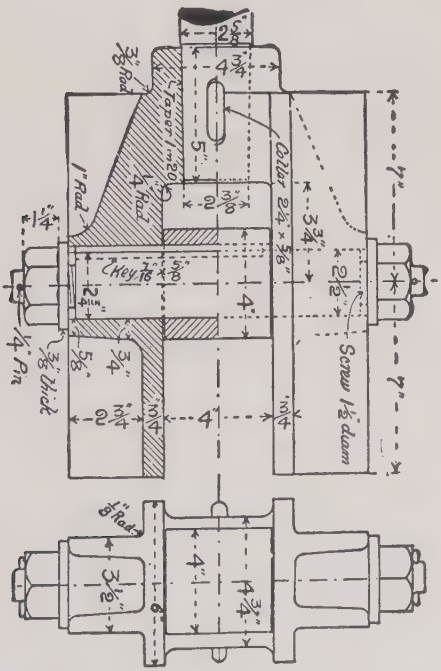


Fig. 203—Locomotive Cross-head

Volume IV is in two parts.

For Index to first part,
see front pages.

For Index to last part,
see end of volume.

Handbook for Millwrights

QUALIFICATIONS.

A millwright, according to Webster, is "one who makes a business of planning and building mills or mill machinery." Therefore in order to become a successful millwright, a man must also be in a certain sense an architect. He should know how to plan, and erect a mill building from foundation to roof truss, in such a manner as will be best adapted to the type of machinery he will be called upon to install. He must be able to calculate strains, the strength of materials, and the resultants of forces. He should be able to make and to read drawings, not only of machinery, but of foundations also, and he should possess technical knowledge sufficient to enable him to make correct calculations regarding the dimensions of foundations, not alone for the buildings, but also for the boilers, engines, and other heavy machines.

He should also be a man full of resources and determination, ready to meet and cope with any emergency or problem which may arise. He should possess the ability to handle men (both skilled and unskilled workmen) in such a manner as to get the best results from their labor. He is frequently called upon to purchase machinery and supplies; therefore he should be a good business man and accountant. The increased, and increasing use of electricity as an agent for the transmission of power in mills and factories has added an-

other to the list of qualifications of the modern millwright, viz.: a good working knowledge of electric machines, such as generators, transformers, motors, rotary converters, electric lights, and the various other electric apparatus with which the modern up-to-date manufacturing establishment is equipped. A complete knowledge of the framing of timber, and the manipulation of structural steel, together with the construction of concrete piers should also be the aim of every man who aspires to become a successful millwright, and he should bear in mind that improvements and changes are continually being made in methods and materials pertaining to his calling. Therefore, he should be a man of study, and should endeavor to make himself thoroughly familiar with all new and improved apparatus, provided it possesses merit. By thus devoting a reasonable portion of his time to reading and study, he will be able to keep step with the march of progress, and always be at the front.

LOCATION OF BUILDINGS.

The location of factory buildings is usually decided upon by the owner, or superintendent, but the millwright is frequently called upon for advice in such cases, and if he is a competent man he will be able to make many valuable suggestions regarding details relating to the best methods of receiving the raw material, in the factory, and the disposal of the finished product. A good rule to follow in designing a manufacturing plant is to see that the course of the material under operation shall, from the time of its reception at the factory, until its exit as a finished product, be in one general direction, viz.; forward in as near a straight

line as possible. It costs money to handle material and any deviation from the above rule will result in increased cost of production. For instance, suppose the building has been so planned, and the machines so located that the material must in the regular course of manufacture be transported a second, or third time to the same floor in order to be worked upon, the added cost of its conveyance, especially if by elevator to a second, or third story, will greatly affect the efficiency of the plant. It is possible to so locate the various machines in the mill in such a manner that the material in its progress need never pass a given point the second time, and this fact should be borne in mind by the millwright in drawing up his plans. Another very pronounced cause of inconvenience and loss of time in doing the work, is the lack of working space around the machines. In the laying out of the different floors, provision should be made for space sufficient around each machine to allow the man in charge of it to operate it without interfering in any way with those next to him.

FOUNDATIONS.

The first thing to be considered in the erection of any structure, is of course the foundation.

The object of foundations is to prevent inequality of settlement and distribute the weight of the structure equally over the substratum.

The bases of structures are invariably made wider than the superincumbent mass, to increase the stability and to counteract all the damaging forces that tend to cause failure.

Damaging Forces.—The principal causes of failure are those which induce settlement, such as inequalities of

earth resistance; the compressibility of mortar joints; lateral escape of soft soil, sliding of the substratum on sloping ground; the withdrawal of water; distributed lateral pressures, causing overturn, such as wind pressure, and thrust of barrel vaulting or of an untied couple raftered roof; concentrated lateral pressure which induces settlement and overturn, such as the thrust of framed floors, trussed roofs and groined vaults subjecting small areas of support to great pressures.

In the case of mill buildings in which heavy machinery is to be operated, there is to be added to the foregoing list of damaging forces, the jar and strain of the moving machinery.

Inequality of Settlement.—Inequality of settlement in buildings takes place from two causes: (1) the compressibility of the mortar joints; (2) the compressibility of the soil.

An allowance of 1 in. in 24 ft. of brickwork in lime mortar is often provided for settlement, as in the example of the extremities of bridging joists of floors, at one end being supported by a brick wall and the other extremities by iron columns, etc.

Nearly all soils, with the exception of solid rock and gravel, are compressible under pressures often attained in buildings. It is therefore impossible, where large buildings are erected on other soils, to avoid settlement; and the fact of any building settling is of no great import, provided the settlement be uniform and of no great depth, and the relative position of the parts of the structure unaltered. But where the resistance of the soil of every part of the site is not uniform, there is a risk of the above defect occurring, and special precautions must be taken to distribute the

pressure to suit the varying strengths of the substratum.

Lateral Escape.—Heavy structures erected upon soft soils, such as running sands and peat, squeeze out from beneath the foundation, unless means are taken to confine the soil to the required area; this is usually accomplished by sheet piling, as described later.

Sliding.—This is a defect usually occurring where the building is erected on the slope of a hill, and the strata inclined, being depressed in the direction and towards the bottom of the slope. The weight of the building is liable to cause the strata to become detached and slide. This is prevented in two ways: (1) by driving piles at intervals to a considerable depth, thus connecting the strata; this method is often objectionable, tending, as it does, to shake and disturb the soil; (2) by building a retaining wall; this is the better method, as it not only supports, but also protects the strata from the effects of the atmosphere, which in soils easily affected by the latter is a desideratum.

Withdrawal of Water from Foundation Earth.—Edifices built on damp soil, such as a sand overlying a clay, have their stability endangered, should the water be drained away after the building has been erected, as it will cause the foundation earth to occupy a less volume and in the sinking will tend to fracture or overturn the walls; therefore the depth of the concrete foundation must be arranged below any probable adjacent cutting.

Distributed Overturning Pressures.—Distributed forces acting upon the upper level of walls, such as the continuous pressure of barrel vaulting, and the spreading tendencies of untied couple raftered roofs, and also the distributed pressures on wall faces, such as wind

pressure, tend to cause failure in two ways: (1) by overturning, the minimum resistance being generally at the change of section, usually at the ground level; (2) by subjecting the leeward edge of the wall to the pressure sufficient to crush the material, or by throwing the weight on a small area of the substratum, forcing it from its original position and causing a settlement.

The safe load which may be placed upon any foundation depends upon the carrying power of the soil and the area of the foundation footing. It is not the purpose of the author to enter into a discussion of the carrying power of the various kinds of soil met with, therefore nothing will be given in that direction save the following table which will serve as a guide to the millwright when he must determine whether or not the plans furnished call for an excessively heavy foundation, or whether the mass of material shown underneath some heavy machine is insufficient for the particular soil met with during the operations at hand.

TABLE 1.—BEARING POWER OF SOILS.
By Prof. Ira C. Baker, University of Illinois.

Kind of Material.	Tons. sq. Ft Min.	Max.
Rock—the hardest—in thick layers, in native bed	200	..
Rock equal to best ashlar masonry	25	30
Rock equal to best brick masonry	15	20
Rock equal to poor brick masonry	5	10
Clay on thick beds (always dry)	4	6
Clay on thick beds (moderately dry)	2	4
Clay, soft	1	2
Gravel and coarse sand, well cemented	8	10
Sand, compact and well cemented	4	6
Sand, clean, dry	2	4
Quicksand, alluvial soils, etc.	0.5	1

Concentrated Lateral Pressure.—The thrust caused by united principals, as groined faults or other forces acting at a point, or along vertical lines on the wall, are often resisted by buttresses.

Atmospheric Action.—Many otherwise thoroughly reliable soils are practically reduced to the condition of mud if exposed to the effects of the atmosphere or to rain water. The variation in temperature at the different seasons also causes the ground to expand and contract considerably.

Where foundations are constructed in such soils, they must be taken sufficiently deep to be beyond the effects of the atmosphere, that is, below the line of saturation.

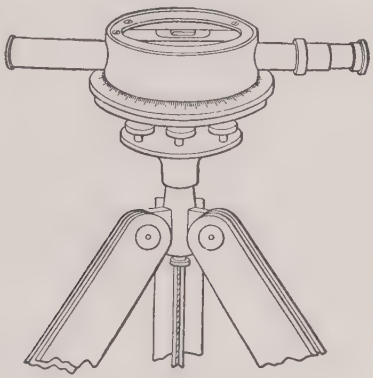


FIG. 1.
SURVEYOR'S LEVEL.

The line of saturation in the section of any part of the earth's crust represents the depth to which the soil at that part is saturated by the absorption of rain water and affected by atmospheric changes.

Excavations.—Before commencing any constructional work in connection with a building, it is necessary as the first operation to carefully take the levels of the site, in order first to arrive at an estimate of the amount of earthwork to be done; and secondly, to determine the design of the basement story, this latter

often being materially affected if the differences in level of the various parts of the site are great. The next operation is to level the ground. This in most instances consists in excavating and removing parts of the site, and in depositing earth in other parts to form embankments or to fill up hollow places. In order to conduct these operations in the most economical manner the levels must in all instances be taken and plotted with the greatest accuracy. This can only be

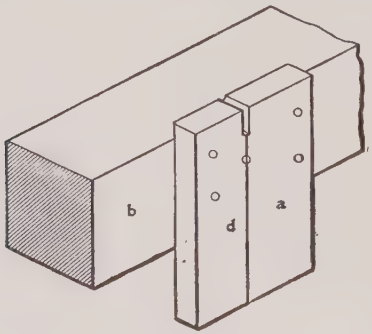


FIG. 2.
TARGET.

efficiently done on areas of any magnitude by means of the surveyor's level, the method of employing which will be described later. All leveling operations for ordinary constructional work may be carried out by referring them to the principles laid down for performing the three following operations:

1. Taking levels of site:

2. Leveling the bottoms of trenches for drains or foundations.

3. Embanking for roads, or leveling of depressions.

Instruments.—The instruments required to determine

the levels of the site are: first, the surveyor's level; second, the measuring staff; third, ranging poles and chain or tape.

Methods of Leveling.—Taking the levels of a site may be carried out in one of three ways: First, by taking a number of section lines across the site; secondly, by

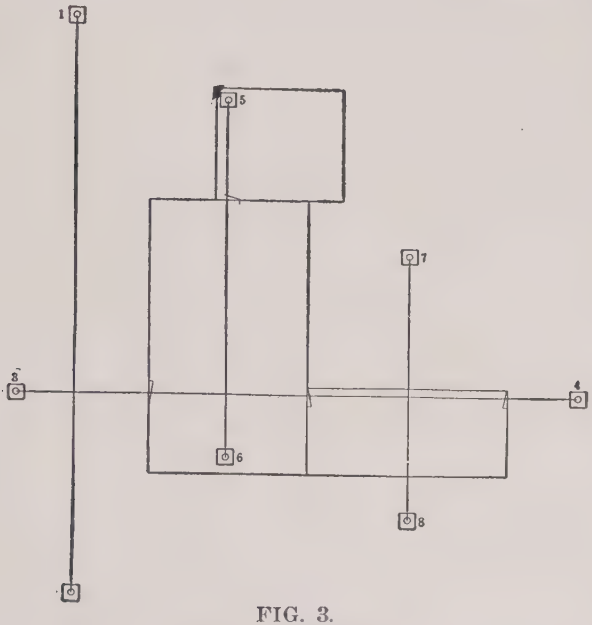


FIG. 3.
PLAN OF STATIONS.

erecting the level in a commanding position and taking the relative heights of the salient points and noting them on plan (this method is only applicable for small sites); thirdly, by contours.

In all three methods it is necessary to have a datum level to commence from, and to which all other levels can be referred. A line on some permanent

structure in the immediate vicinity is usually taken, or if such does not exist, a stout stake is driven in the ground in a position away from the work where it is not likely to be disturbed.

First Method.—A number of sections are ranged across the site, each line being numbered or lettered; the level is then set up on, or in close proximity to the first line and the datum; the measuring staff is then held by an assistant on the datum point and then on

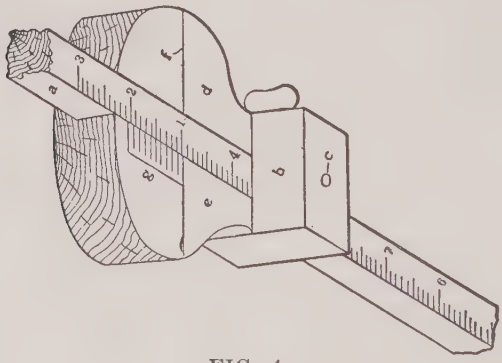


FIG. 4.
LEVELING ROD.

the extremity of the line, the relative heights of the two points being recorded in a field book kept for that purpose. A number of points on the line are then taken, and the measuring staff is held over them and their relative heights are recorded, and their distances from the beginning of the line are measured. When the bottom of the measuring staff rises above, or the top becomes depressed below the line of sight through the rise or depression of the ground, the level must be moved further along the line and the preceding

operations repeated. Fig. 5 illustrates the method. The following is a form of field book with the reading for a section entered:

FIELD LEVEL BOOK.

Back Sight.	Inter. Sight.	Fore-Sight.	Rise	Fall.	Reduced Levels.	Distance.	Total Distance.	Remarks.
						chains		
4.15	.....	.....	.....	....	100.0	.....	.....	Bench Mark A
	4.13	.....	.02	....	100.02	1	.....	1 peg
	5.01	.....	.....	.88	99.14	2	.....	2
	4.86	.....	.15	....	99.29	3	.....	3
	6.06	.....	.....	1.20	98.09	4	.....	4
		8.02	.....	1.96	96.13	5	.....	5
12.25								
	8.46	.....	3.79	....	99.92	6	.....	6
	3.04	.....	5.42	....	105.34	7	.....	7
12.60								
		2.15	.89	....	106.23	7.57	.....	Bench Mark B
	7.19	.....	5.41	....	111.64	8.57	.....	8
		2.53	4.66	....	116.30	9.57	.....	9
9.37								
	5.75	.....	3.62	....	119.92	10.57	.....	10
		3.94	1.81	....	121.73	11.57	11.57	Bench Mark C
			25.77	4.04	21.73			
			4.04					
			21.73					

The above shows a typical field book. The reduced level of the first point is taken as 100 ft. above a datum level; the levels are all read in feet and hundredths of a foot; the distances are taken in chains and links, but may be taken in feet and inches. The rise and fall columns should be balanced, also the first and last reading in the reduced levels; these two quantities will equal each other if the computations have been correctly made.

Second Method.—The second method is evident from the previous explanations.

Third Method.—The method of contouring is the most useful, but takes the longest time to perform it: it consists in describing upon a plan a series of level lines with a uniform vertical interval between them.

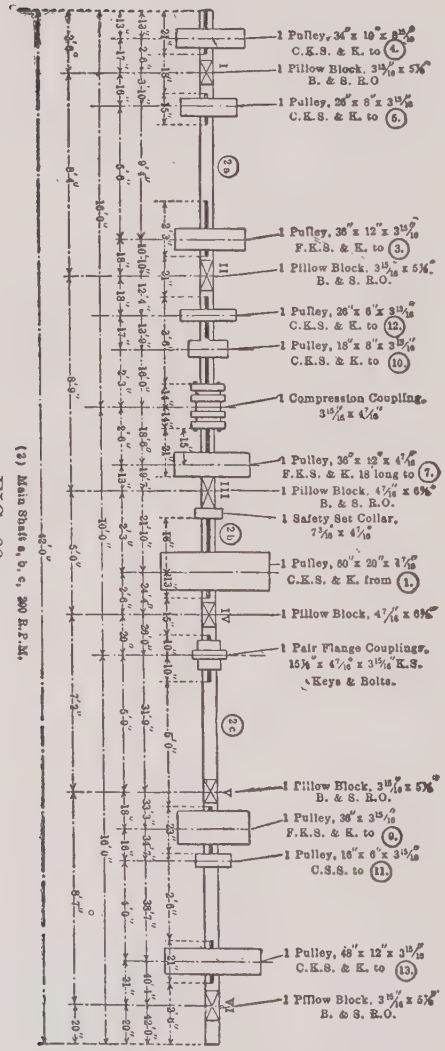

1 Pulley, 34" x 10" x 3 15/16"
C.K.S. & K. to 4.
1 Pillow Block, 3 15/16" x 5 7/8"
B. & S. R.O
1 Pulley, 26" x 8" x 3 15/16"
C.K.S. & K. to 5.
1 Pulley, 36" x 12" x 3 15/16"
F.K.S. & K. to 3.
1 Pillow Block, 3 15/16" x 5 7/8"
B. & S. R.O.
1 Pulley, 26" x 6" x 3 15/16"
C.K.S. & K. to 12.
1 Pulley, 18" x 8" x 3 15/16"
C.K.S. & K. to 10.
1 Compression Coupling,
3 15/16" x 4 7/16"
1 Pulley, 36" x 12" x 4 7/16"
F.K.S. & K. 18 long to 7.
1 Pillow Block, 4 7/16" x 6 5/8"
B. & S. R.O.
1 Safety Set Collar,
7 3/16" x 4 7/16"
1 Pulley, 60" x 20" x 3 7/16"
C.K.S. & K. from 1.
1 Pillow Block, 4 7/16" x 6 5/8"
1 Pair Flange Couplings,
15 1/8" x 4 7/16" x 3 15/16" K.S.
Keys & Bolts.
1 Pillow Block, 3 15/16" x 5 7/8"
B. & S. R.O.
1 Pulley, 36" x 3 15/16"
F.K.S. & K. to 9.
1 Pulley, 18" x 6" x 3 15/16"
C.S.S. to 11.
1 Pulley, 48" x 12" x 3 15/16"
C.K.S. & K. to 13.
1 Pillow Block, 3 15/16" x 5 7/8"
B. & S. R.O.
(2) Main Shafts a, b, c, 200 R.P.M.

FIG. 66.

on a scale of ½ inch, or 1 inch to the foot, the latter scale to be used when there are many pulleys or gear wheels on a shaft. The dimensions of the drawings should be about 24 x 36 inches, and several shafts can be placed on the same sheet. The shaft manufacturer requires a drawing that shows plainly each and every part that is to go on the shaft, and its exact location.

Fig. 66 is a reproduction of a drawing for the main-line shaft of a shop equipped with a 125 H. P. engine belted to the 60 x 20-inch pulley shown near the middle of the shaft. The engine shaft is roughly located at [1]. The line shaft is indicated by [2], and the drawing is marked "[2] Main Shaft a, b, c, 200 R. P. M." The number in the brackets refers to the corresponding number on other drawings of the building which may chance to show the main-line shaft. It is also found in the different lists of parts, as will be seen. First at bottom comes the dimension over all line, showing the total length of the shaft. The second line gives the exact distances apart (center to center) of the bearings, a good guide for the millwright when erecting the bridge trees. The fifth line from the bottom, which is also the second from the shaft, gives the dimensions from each pulley center to ends of shaft, to centers of bearings, and these dimensions having been obtained by scaling the drawing with great accuracy, serve as a check by which each of the other lines of dimensions may be verified. The dimensions in the other lines are obtained from this line. A beginning should be made at the end of the shaft most convenient, and the bearing nearest to that end is marked I on the drawing.

The next bearing is marked II, and the others III, IV, etc., to the other end of the shaft. The end of the shaft next to bearing I should always be taken as a

starting point in making other drawings or designs that are in connection with this particular line of shafting. Referring to Fig. 66 and starting at the left-hand end of the shaft it will be seen that the distance to the first center on the shaft, that of a pulley is 13 inches. Mark this down in its place on the second line from the shaft, and place witness marks on points or lines between which measurement was made. The next dimension point is the center of a bearing, and this distance is also scaled from the end of the shaft from which the first measurement was made. The distance, 2 feet 6 inches, is marked on the line, as close as possible to the bearing center line, and another witness mark placed on this line. It should be remembered that all dimensions marked on this second line are the distances of the various centers from one common fixed point, viz., the witness mark on the line coincident with the end of the shaft.

The measurements are continued in this manner and the distances entered in their proper places, until the other end of the shaft is reached. The total length, or over-all dimension of the shaft is found to be 42 ft., which is entered between the two last witness marks. The millwright thus has before him at a glance, and without any adding of other dimensions, the over-all dimensions of the shaft, and in order to obtain the distances between centers of pulleys, or from centers of pulleys and centers of bearings, in fact, any local dimensions on the shaft, all that is required is to subtract from each other the scaled dimensions of the two local points as they appear on the over-all dimensions line, and the result is the distance between the two points.

As an example, suppose it is desired to obtain

the distance between bearings I and II; upon this line, near the center line of bearing I appears the notation, 2 feet 6 inches, and near bearing II appears 10 feet 10 inches. Deducting the former from the latter leaves a remainder of 8 feet 4 inches, which is the required distance between the centers of the two bearings. In this manner the local distance between any two points on the shaft may be easily ascertained. The line of dimensions nearest to the shaft as it appears in Fig. 66 has upon it a row of figures for guidance in cutting the key-ways. This line is for the special use of the machinist, and when the correct dimensions are placed upon it and strictly adhered to in the work, very little trouble will be experienced in placing the pulleys. In the drawing these key-way dimensions are either tied to the end of each piece of shaft, or to each other, and in laying out the splining, work can be started at one end and prosecuted as far as dimensions appear; when it can be started at the other end and completed.

Again referring to Fig. 66, it will be noticed that at the top there is a list containing names, dimensions, and descriptions of each and every piece of machinery to be placed upon the shaft. If any particular piece has a connection with other shafting a notation is also made of this. In assembling a shaft a drawing of this character will be found to be of inestimable service. By reference to this list the erector can see at a glance a description of each part. For instance take the pulley located near pillow block III; the notation on the drawing, opposite this pulley is, 1 pulley 36″ x 12″ x 4 7/16″ F, K, S, and K, 18″ long, which being interpreted, means that the pulley is 36 inches in diameter, 12 inches face and the hub is bored 4 7/16 inches, flat faced, key-seated and key to be 18 inches long.

Where the letters C, S, S, occurr, the pulley is to be crown faced, set screwed. The pillow-blocks are marked B and S, R, O, thus indicating that they are ball and socket, ring or chain oiling, as the case may be. In order to distinguish these from the solid bearings, the latter are simply marked 1 pillow block, or solid flat box. It will thus be seen that by reference to the drawing

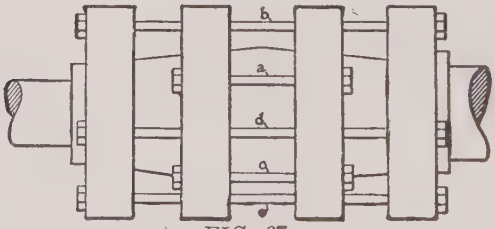


FIG. 67.
CORRECT ARRANGEMENT OF BOLTS IN COMPRESSION COUPLING.

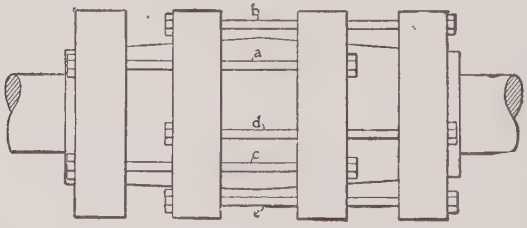


FIG. 68.
INCORRECT ARRANGEMENT OF BOLTS IN COMPRESSION COUPLING.

the workman need never to be in doubt concerning the correct location of any particular part.

Compression Couplings.—Figs. 67 and 68 show two kinds of compression couplings. The coupling shown in Fig. 67 is without doubt the most reliable. The alternate bolt arrangement in this coupling while it securely holds the parts together, at the same time permits of greater latitude in the twisting of the shaft

when placed under heavy strain. In fact, with a coupling of this character properly made, and placed, the probabilities are that the shaft would be twisted in two at some other point before the coupling would slip. But with an arrangement of bolts similar to that shown in Fig. 68, a coupling frequently lets go, due to the fact that the rings which clamp the heel and toe of the split compression cone are not connected together as in Fig. 67, but the heel rings or collars are simply connected to each other, likewise the two toe collars. It will thus be seen that the pressures are not equalized as they are by the arrangement shown in Fig. 67.

In the drawing, Fig. 66, it will be noted that one compression coupling, and one flange coupling are shown. This is for the purpose of illustrating the two methods of connecting the sections of shaft. In practice it is customary to have the couplings all of one type. Referring to Fig. 66, the flange coupling will be found near pillow block IV, and a compression coupling is shown between pillow blocks II and III.

Horse Power of Shafting.—Table 17 will be found a valuable aid in ascertaining the horse power of shafting of various dimensions, and running under different conditions. It shows at a glance the horse power that the shaft will transmit when running at various speeds.

Table 18 gives the dimensions of keys and thrust collars, and should be used in connection with the drawings at top in which the lettering corresponds with that of table 18.

TABLE 17

HORSE-POWER OF SHAFTING.

AS MAIN SHAFTING, CARRYING DRIVING PULLEYS OR GEARS.

Diameter of Shaft in Inches.	Number of Revolutions per Minute.										
	60	80	100	125	150	175	200	225	250	275	300
1⅜	2·6	3·4	4·3	5·4	6·4	7·5	8·6	9·7	10·7	11·8	12·9
2	3·8	5·1	6·4	8	9·6	11·2	12·8	14·4	16	17·6	19·2
2⅛	5·4	7·3	8·1	10	12	14	16	18	20	22	24
2¼	7·5	10	12·5	15	18	22	25	28	31	34	37
2⅜	10	13	16	20	24	28	32	36	40	44	48
3	13	17	20	25	30	35	40	45	50	55	60
3⅛	16	22	27	34	40	47	54	61	67	74	81
3½	20	27	34	42	51	59	68	76	85	93	102
3⅝	25	33	42	52	63	73	84	94	105	115	126
4	30	41	51	64	76	89	102	115	127	140	153
4½	43	58	72	90	108	126	144	162	180	198	216
5	60	80	100	125	150	175	200	225	250	275	300
5½	80	106	133	166	199	233	266	299	333	366	400

AS SECOND MOTION OR LINE SHAFTING.

BEARINGS EIGHT FEET APART.

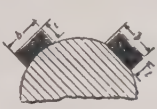
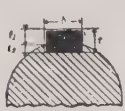
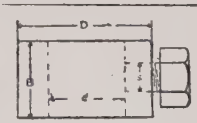
Diameter of Shaft in Inches.	Number of Revolutions per Minute.										
	100	125	150	175	200	225	250	275	300	325	350
1⅜	6	7·4	8·9	10·4	11·9	13·4	14·9	16·4	17·9	19·4	20·9
1⅞	7·3	9·1	10·9	12·7	14·5	16·3	18·2	20	21·8	23·6	25·4
2	8·9	11·1	13·3	15·5	17·7	20	22·2	24·4	26·6	28·8	31
2⅛	10·6	13·2	15·9	18·5	21·2	23·8	26·5	29·1	31·8	34·4	37
2¼	12·6	15·8	19	22	25	28	31	35	38	41	44
2⅜	15	18	22	26	29	33	37	41	44	48	52
2½	17	21	26	30	34	39	43	47	52	56	60
2¾	23	29	34	40	46	52	58	64	69	75	81
3	30	37	45	52	60	67	75	82	90	97	105
3⅛	38	47	57	66	76	85	95	104	114	123	133
3½	47	59	71	83	95	107	119	131	143	155	167
3¾	58	73	88	102	117	132	146	162	176	190	205
4	71	89	107	125	142	160	178	196	213	231	249

FOR SIMPLY TRANSMITTING POWER.

Diameter of Shaft in Inches.	Number of Revolutions per Minute.										
	100	125	150	175	200	233	267	300	333	367	400
1½	6·7	8·4	10·1	11·8	13·5	15·7	17·9	20·3	22·5	24·8	27
1⅝	8·6	10·7	12·8	15	17·1	20	22·8	25·8	28·6	31·5	34·3
1¾	10·7	13·4	16	18·7	21·5	25	28	32	36	39	43
1⅞	13·2	16·5	19·7	23	26·4	31	35	39	44	48	52
2	16	20	24	28	32	37	42	48	53	58	64
2⅛	19	24	29	33	38	44	51	57	63	70	76
2¼	22	28	34	39	45	52	60	68	75	83	90
2⅜	27	33	40	47	53	62	70	79	88	96	105
2½	31	39	47	54	62	73	83	93	104	114	125
2¾	41	52	62	73	83	97	111	125	139	153	167
3	54	67	81	94	108	126	144	162	180	198	216
3⅛	68	86	103	120	137	160	182	205	228	250	273
3½	85	107	128	150	171	200	228	257	285	313	342

TABLE 18

DIMENSIONS OF KEYS AND THRUST COLLARS.

Diameter of Shaft in Inches.	Keys.					Cast-Iron Collars.				Diameter of Shaft in Inches.
										
d	b	t	t	t_1	t_2	D	B	s	No. of Screws.	d
1	$\frac{3}{8}$	$\frac{5}{32}$	$\frac{1}{4}$	$\frac{5}{32}$	$\frac{5}{32}$	$1\frac{3}{4}$	1	$\frac{3}{8}$	1	1
$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{5}{32}$	$\frac{7}{16}$	$\frac{5}{32}$	$2\frac{1}{8}$	1	,,	,,	$1\frac{1}{4}$
$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{32}$	$\frac{7}{16}$	$\frac{5}{32}$	$\frac{5}{32}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{2}$	,,	$1\frac{1}{2}$
$1\frac{3}{4}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{11}{32}$	$\frac{5}{32}$	$\frac{1}{8}$	$2\frac{7}{8}$	$1\frac{7}{16}$	,,	,,	$1\frac{3}{4}$
2	$\frac{5}{8}$	$\frac{5}{32}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$3\frac{1}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	,,	2
$2\frac{1}{4}$	$\frac{11}{16}$	$\frac{5}{16}$	$\frac{13}{32}$	$\frac{1}{4}$	$\frac{5}{32}$	$3\frac{1}{2}$	$1\frac{7}{8}$	,,	,,	$2\frac{1}{4}$
$2\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{5}{32}$	$\frac{5}{32}$	$3\frac{3}{4}$	2	,,	,,	$2\frac{1}{2}$
$2\frac{3}{4}$	$\frac{13}{16}$	$\frac{11}{32}$	$\frac{7}{16}$	$\frac{5}{32}$	$\frac{5}{32}$	$4\frac{1}{8}$	$2\frac{3}{16}$	,,	,,	$2\frac{3}{4}$
3	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{7}{16}$	$4\frac{3}{8}$	$2\frac{3}{8}$	$\frac{3}{4}$	,,	3
$3\frac{1}{4}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{7}{16}$	$4\frac{3}{4}$	$2\frac{1}{2}$	,,	,,	$3\frac{1}{4}$
$3\frac{1}{2}$	1	$\frac{13}{32}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$5\frac{1}{8}$	$2\frac{3}{4}$	,,	,,	$3\frac{1}{2}$
$3\frac{3}{4}$	$1\frac{1}{16}$	$\frac{13}{32}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$5\frac{1}{2}$	$2\frac{7}{8}$	,,	,,	$3\frac{3}{4}$
4	$1\frac{1}{8}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{13}{32}$	$\frac{5}{32}$	$5\frac{3}{4}$	3	,,	2	4
$4\frac{1}{4}$	$1\frac{3}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{13}{32}$	$\frac{5}{32}$	$6\frac{1}{8}$	$3\frac{1}{4}$	,,	,,	$4\frac{1}{4}$
$4\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{7}{16}$	$\frac{1}{4}$	$6\frac{1}{2}$	$3\frac{1}{2}$	,,	,,	$4\frac{1}{2}$
$4\frac{3}{4}$	$1\frac{5}{16}$	$\frac{7}{16}$	$\frac{11}{16}$	$\frac{7}{16}$	$\frac{1}{4}$	$6\frac{3}{4}$	$3\frac{5}{8}$	,,	,,	$4\frac{3}{4}$
5	$1\frac{3}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$7\frac{1}{8}$	$3\frac{3}{4}$	,,	,,	5
$5\frac{1}{4}$	$1\frac{7}{16}$	$\frac{9}{32}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$7\frac{1}{2}$	4	,,	,,	$5\frac{1}{4}$
$5\frac{1}{2}$	$1\frac{1}{2}$	$\frac{9}{32}$	$\frac{13}{16}$	$\frac{7}{16}$	$\frac{5}{32}$	$7\frac{3}{4}$	$4\frac{1}{8}$	,,	,,	$5\frac{1}{2}$
$5\frac{3}{4}$	$1\frac{9}{16}$	$\frac{9}{32}$	$\frac{13}{16}$	$\frac{7}{16}$	$\frac{5}{32}$	$8\frac{1}{8}$	$4\frac{1}{4}$	,,	,,	$5\frac{3}{4}$
6	$1\frac{5}{8}$	$\frac{5}{16}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{5}{32}$	$8\frac{1}{2}$	$4\frac{1}{2}$	,,	,,	6
$6\frac{1}{4}$	$1\frac{11}{16}$	$\frac{11}{32}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{5}{32}$	$8\frac{3}{4}$	$4\frac{5}{8}$	$\frac{7}{8}$	,,	$6\frac{1}{4}$
$6\frac{1}{2}$	$1\frac{3}{4}$	$\frac{11}{32}$	$\frac{15}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	9	$4\frac{3}{4}$	,,	,,	$6\frac{1}{2}$
$6\frac{3}{4}$	$1\frac{13}{16}$	$\frac{3}{8}$	$\frac{15}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$9\frac{3}{8}$	5	,,	,,	$6\frac{3}{4}$
7	$1\frac{7}{8}$	$\frac{3}{8}$	1	$\frac{11}{16}$	$\frac{7}{16}$	$9\frac{3}{4}$	$5\frac{1}{4}$	,,	,,	7
$7\frac{1}{4}$	$1\frac{15}{16}$	$\frac{13}{32}$	1	$\frac{11}{16}$	$\frac{7}{16}$	10	$5\frac{3}{8}$	,,	,,	$7\frac{1}{4}$
$7\frac{1}{2}$	2	$\frac{13}{32}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{7}{16}$	$10\frac{3}{8}$	$5\frac{1}{2}$	,,	,,	$7\frac{1}{2}$
$7\frac{3}{4}$	$2\frac{1}{16}$	$\frac{7}{16}$	$1\frac{1}{16}$	$\frac{3}{4}$	$\frac{7}{16}$	$10\frac{3}{4}$	$5\frac{3}{4}$	,,	,,	$7\frac{3}{4}$
8	$2\frac{1}{8}$	$\frac{7}{16}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{8}$	$11\frac{1}{4}$	6	,,	,,	8

Erection of Shafting.—When it is necessary to erect piers for supporting a line of shafting, whether the piers be of concrete, stone, or brick, one point should never be lost sight of, viz: that in addition to the dead weight of the shafting and pulleys, there will also be the strains caused by the pull of the belts, some in one direction and some in another, and these strains do not come on all the piers alike, some being called upon to withstand more than others. Therefore, the proper thing to do is to design each pier with a view to the strain that is likely to come upon it. For instance, let

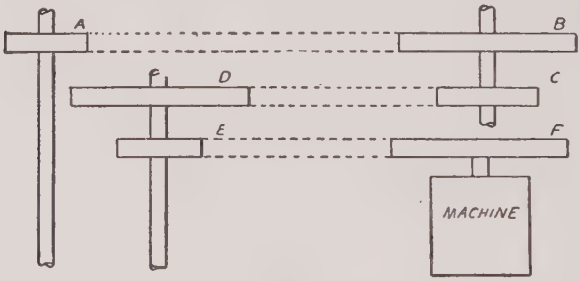


FIG. 70
REDUCING PULLEY SPEED IN SEVERAL STEPS.

Fig. 71 represent a line of shafting located 6 ft. above the floor line, and it is required to ascertain the dimensions and shape of each of the several piers, marked I, II, III, V and VI.

For present purposes the length of the shaft is assumed to be 34 ft., and the diameter is $3\frac{15}{16}$ inches. The weight of the shaft and pulleys being known, it remains to ascertain the strains that will result from the pull of the belts.

It is customary to limit the force of this pull to 400 lbs. per inch width of belt, and by reference to Fig. 71, it will be seen that there is a pulley D in close

proximity to pier VI. The width of the belt running over this pulley is 14 inches. There will therefore be $400 \times 14 = 5600$ lbs. exerted in a horizontal direction at a height of 6 feet from the floor line, illustrated at a, Fig 72. Should it be necessary to go to a depth of 4 feet in order to secure a suitable bearing soil for the foot of the pier, the force of 5,600 lbs. would be exerted through a lever $6 + 4 = 10$ feet in length, and as the

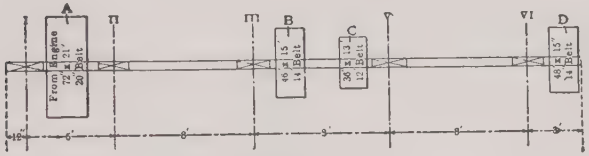


FIG. 71.
SHAFT SUPPORTING PIERS.

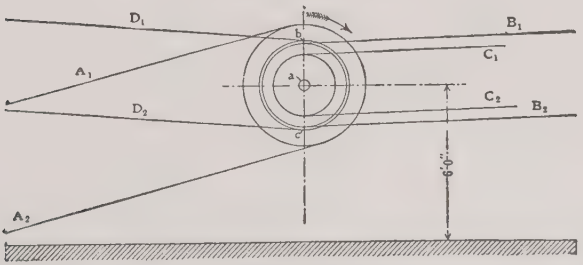


FIG. 72.
BELT PULL ON SHAFT SUPPORTING PIERS.

force in pounds applied at the end of a lever multiplied by the length of the lever in feet reduces the leverage to foot pounds of statical moment, the moment of the force of 5600 lbs. will be $5600 \times 10 = 56000$ lbs.

To resist this force, two factors are to be considered; first, weight of the pier, and second length of foundation at bottom. This is illustrated in Fig. 73 to which reference is now made. If the foundation there shown

was only one foot in length at its base and weighed 56000 lbs., it would just balance the pull of the belt, and if it were made two feet long, as from d to f for instance, its weight could be reduced one-half, or $56{,}000 \div 2 = 28{,}000$ lbs. and it would still balance the belt pull, provided the weight could all be applied at d, which cannot be done because the weight (28000 lbs.) is distributed over the whole area of the base extending to point f, which is in fact the fulcrum.

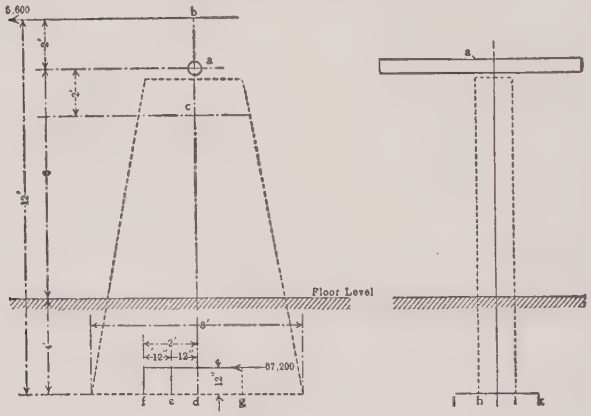


FIG. 73.
CALCULATING WEIGHT NECESSARY IN A PIER.

For this reason the actual holding down weight or power of the pier in its present form is only one-half what it appears to be, and it becomes necessary to increase its length to point g, Fig. 73, thus making it 4 feet in length at right angles to the shaft and an effective weight of 28000 lbs.

The width of the pier parallel with the shaft depends upon the nature of the soil or footing, and the total weight required. The figures given above merely show the net weight required to balance the pull of the belt

on pier VI (Fig. 71), assuming this pull to be exerted at a, Fig. 73, but as pulley D is 4 feet in diameter, there should be one-half of this diameter added to the length of the lever, making it 12 instead of 10 feet, in length, and with a moment equal to $5600 \times 12 = 67200$ lbs., and the actual holding down weight required from

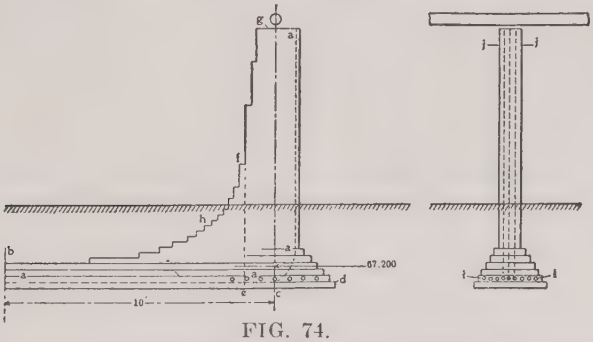


FIG. 74.
ECONOMY OF REINFORCED CONCRETE.

d to g will be $67,200 \div 2 = 33,600$ lbs. From this may be deducted the combined weights of:

7 feet of shaft, $3\frac{15}{16}$ inches in diam.	288 lbs.
The pillow block	100 lbs.
Four bolts	25 lbs.
1 Pulley 48 x 15 inches	550 lbs.
30 feet of 14 inch 6 ply belt	140 lbs.
Total	1103 lbs.

The weight of masonry necessary will then be $33600 - 1103 = 32497$ lbs. The weight of the material to be used will range from 140 lbs. per cu. ft. for brick work, to 160 lbs. per cu. ft. for concrete. If the pier be built of brick, there will be required $32497 \div 140 = 232$ cu. feet of masonry. As the pier is to be

10 feet high, it is obvious that the cross-sectional area must be $232 \div 10 = 23.2$; and if the pier is to be made 4 feet long, its width must be about 5 feet 10 inches, or wider than it is long.

It is obvious that the above noted dimensions are not very suitable for the pier in question. Therefore, the length may be increased to 8 feet, which will double the

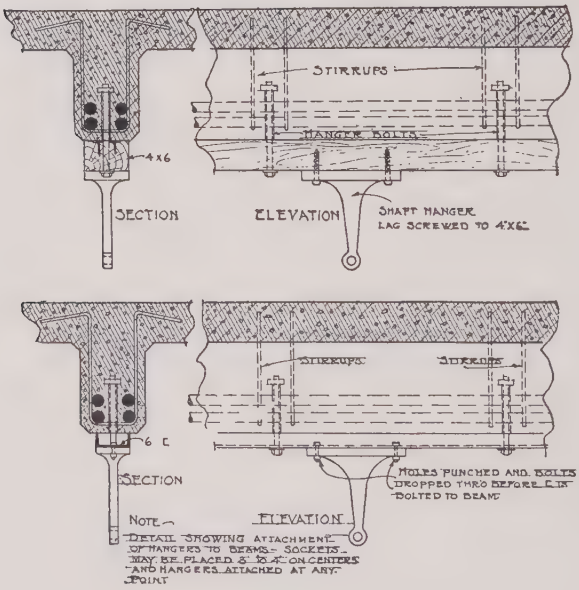


FIG. 75.

turning resistance, while at the same time the holding down weight may again be reduced one-half or $33600 \div 2 = 16800$ lbs. Deducting the weight of shaft, pulley, etc., from this, will leave $16,800 - 1,103 = 15,697$ lbs. of masonry required, equal to $15697 \div 140 = 112$ cu. feet.

If the pier be made 8 feet in length at the foot, and

tapering to 4 feet in length at the top, the average length would be 6 feet, giving a surface area of $10 \times 6 = 60$ square feet, and as 112 cu. feet of masonry are required, the width should be $112 \div 60 = 1\ 5/6$ feet or say 2 feet.

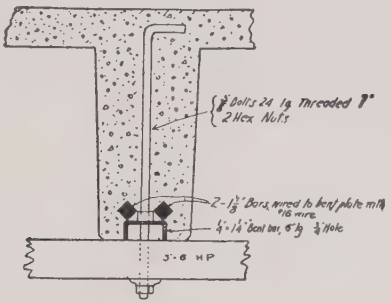


FIG. 76.

The foot of the pier has an area of $8 \times 2 = 16$ sq. feet, and as the load is about 8 tons, there will be one-half ton per sq. foot which is a safe load without any extension of footing.

Both side and end elevation are outlined in Fig.

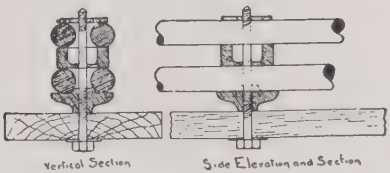


FIG. 77.

73. Another type of pier may be constructed as shown in Fig. 74, in which the quantity of masonry is reduced from 112 to 6 cubic feet. In this form of construction, the bearing for the 67,200-pound thrust moment 12 inches above the bottom of the foundation is

carried 10 feet away toward b, and instead of having been able to divide the thrust moment by 4, as in Fig. 73, we are now able to divide it directly by 10, the footing at b, and intermediate between b and c being sufficient to absorb the pressure.

The division by 10 allows the belt pull to be balanced by a masonry load of 6720 pounds, and deducting the shaft-load of 1103 pounds there remains only 5617 pounds weight of masonry necessary, or a trifle more than 40 cubic feet for the pillar g, e.

The cross section will now be $40 \div 10 = 4$ sq. feet, which corresponds to its being 4 feet long and 12 inches

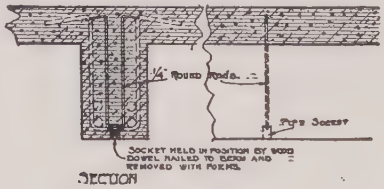


FIG. 78.

thick. This takes no account of the brace that extends to b, more than half of which in weight is accountable as being attached to g, e, thereby increasing the resistance of the pier. There is also the holding power of the soil which may be taken into account if necessary.

In order to give the required strength to withstand the transverse or breaking strain due to the pull of the belt, the pier is reinforced with steel bars placed as shown at a, a, a, a, Fig. 74. The strength of steel necessary at any point between b and c may be found by considering its distance from c as the length of a lever which is to be divided into the moment—67,200. Thus at a point 6 feet from c, toward b, the load will be

$67{,}200 \div 6 = 11{,}200$ pounds, which may be considered as transverse strain tending to break the brace foundation at a point 6 feet from c.

ATTACHING SHAFTING AND MACHINERY IN REINFORCED CONCRETE BUILDINGS.

The following very able discussion of this most important subject is presented by Wm. M. Bailey, M. A. M.

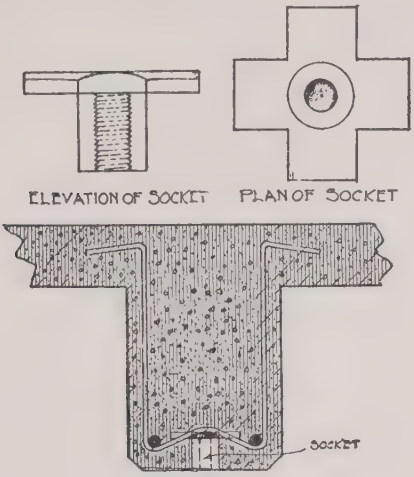


FIG. 79.

Soc. C. E., and Chief Engineer, Eastern Concrete Construction Co., Boston, Mass.:

The manner of attaching shaft hangers, pipes and machinery to beams and floors in a factory or mill building of the reinforced concrete type, is a detail of considerable importance, so much so in fact, that often if no satisfactory scheme is explained to the owner, he will not consider the use of reinforced concrete beams in his proposed building. In the ordinary mill con-

structed factory with wooden or steel floor beams, there is no special difficulty in attaching hangers for shafting and pipes, or fastening machinery to the beams and floors in any desired location without any previous plans necessary and at a minimum cost. But in a concrete building the conditions are entirely different, and it is much better to adopt some scheme for the installation of the machinery, piping, etc., before the building is constructed if we wish to obtain the best results in all respects.

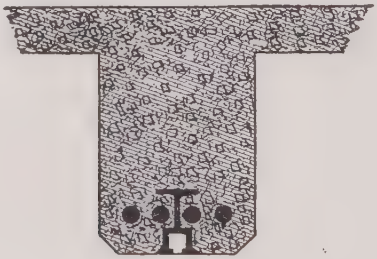


FIG. 80.

Shaft Hanger Supports.—Appliances for conveniently supporting hangers for pipes and shafting, and for supporting motors and other machinery from the beams and ceiling, probably require more previous planning and careful selection than is necessary for other phases of this subject. The method recommended by the engineer and adopted for use with this type of building, should be one designed to best meet all the requirements so far as can be known of the building and its later equipment. It should be inexpensive, compact and simple in form, with not too many separate parts, and as flexible as possible to accommodate future changes in the line of shafting, additions or improved machinery, etc.

One of the most common methods employed, is to attach a hard pine timber to the under side of the concrete beam. This may be in any desired length, to which the shaft hangers are bolted or lag-screwed in their exact position, as is shown in Fig. 75. The timber can be secured to the bottom of the beams by bolts

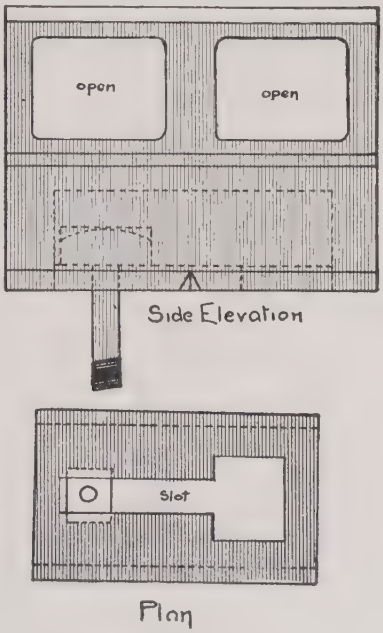


FIG. 81.

built into the concrete about 4 ft. on centers with a projecting threaded end and nut below, or by the use of sockets, inserts, sleeves and washer nuts imbedded in the concrete, into which the timber pieces are afterwards fastened. If the bolts are used, care should be taken to set and keep them vertical, also have them

thoroughly anchored into the upper part of the beams.

Fig. 76 illustrates the long bolts with saddles and check nuts to hold them in position, which is quite essential. It is seldom that all the bolts projecting from the bottom of the beam are made use of and where not used they present an unsightly appearance, hence sockets or sleeves and washer nuts are often preferable. These can be kept in correct position with less trouble,

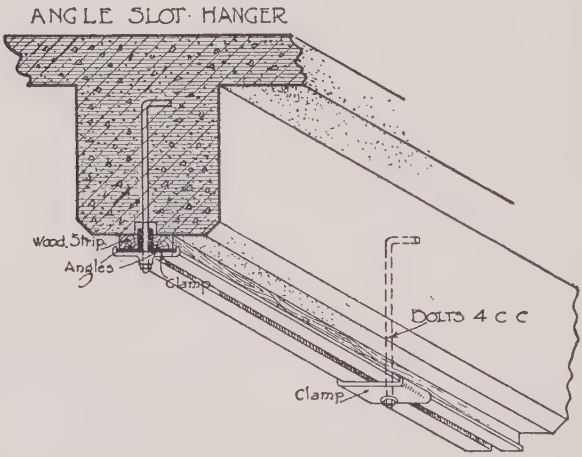


FIG. 82.

at about the same cost and surely make a neater looking job.

In Fig. 77 we have shown the Unit socket, the well known insert for concrete work. This device serves also as a support and spacer for the beam rods, keeping them all properly spaced from each other and from the beam box or centering.

Fig. 78 represents a socket sometimes used. It is made of a short piece of wrought iron pipe tapped out on the inside to receive a lag screw or bolt. Holes are

drilled near the top through which a ¼-inch rod or stirrup is placed which should extend well up into the concrete. The socket itself is temporarily held in position before the concrete is placed, by setting it over a

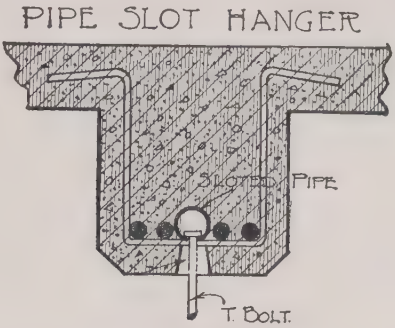


FIG. 83.

wooden plug nailed to the centering. The stirrups are not easily kept in an upright position. For this reason, this socket is not altogether satisfactory.

The socket shown in Fig. 79 is made of cast, or mal-

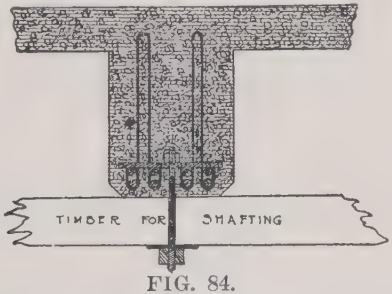


FIG. 84.

leable iron, with a flanged top which serves to anchor it into the concrete. A vertical stirrup should be used with this socket if it is placed in a beam, and is intended to support much load. If used as shown in Fig.

79, it also acts as a support for the rods. The Bigelow socket shown in Fig. 80 can be made use of for almost all supports in reinforced concrete work.

It is a malleable casting made in many sizes, can be

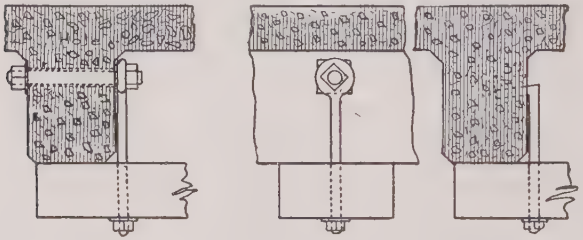


FIG. 85.

anchored as securely in the concrete as may be necessary by the use of stirrups, and is more or less adjustable. Another form of adjustable sockets or inserts for use in beams principally is shown in Fig. 81. These

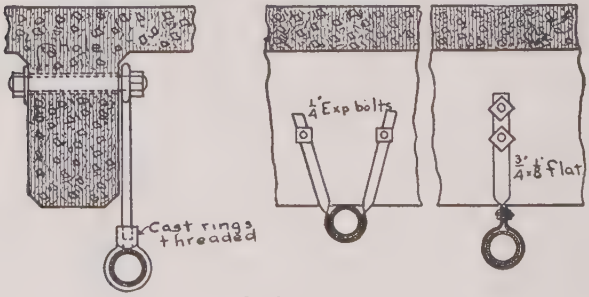


FIG. 86.

castings are made in convenient lengths, and are set on the forms for centering before the concrete is placed. The slot in the bottom makes it possible to place hangers, or bolts at any desired location along the length of the insert. Fig. 75 illustrates the use of large

washer nuts imbedded in the upper part of the concrete beam, usually about 4 feet on centers. These are temporarily held in position by a thin iron tube resting on the centering, and by a bolt projecting through the hole in the bottom of the beam box. The washer nut is tightened down to the top of the tube which secures it firmly in an upright position. When the beam boxes or centering are taken down, the bolts are easily unscrewed from the nuts, thus leaving a clear

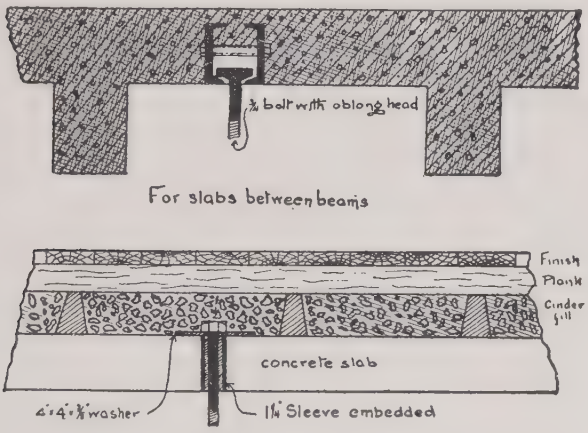


FIG. 87.
METHODS USED FOR SLABS BETWEEN BEAMS.

passage through the concrete to the nut above, to which can be attached timber, or iron stringer onto which the shaft hangers can be fastened in the ordinary manner.

Figs. 82 and 83 illustrate two other methods used for attaching shaft hangers and machinery to beams. With either of these schemes the shaft hangers or other fittings may be bolted or clamped to the concrete structure without the use of wood timbers, or iron bolsters,

if so desired. The method shown in Fig. 82 is one that was adopted by the United Shoe Machinery Company, in its extensive shops at Beverly, Mass.

Another method is suggested in Fig. 84, which may be made a strong and serviceable support for motors and other machinery. Other systems are shown in Figs. 85 and 86. When a bolt is used through the beam near the top, a metal sleeve having a wooden core is con-

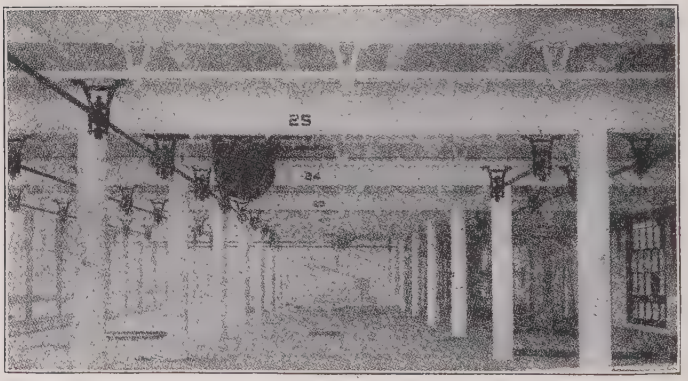


FIG. 88.
GENERAL VIEW OF SHAFTING.

creted around to form the hole. These openings also permit the passage of pipes and wires.

Large holes left in certain parts of beams and girders might cause weakness in the compressive side of the member. In such cases, an iron pipe sleeve of sufficient thickness should be used. It is rather difficult to always locate the holes through the beams during construction, so as to conform exactly with future changes. Therefore, the system using the hooked hanger bolt inside the beam is the most satisfactory. If the holes

for the hook bolts are cut with a pneumatic drill, it can be done at a nominal cost.

Wires and Sprinkler Supports.—Blocks of wood are often set in the under side of concrete slabs to which are fastened wires, and small sized pipes, and wire conduits. This is not considered a suitable, or permanent support. It is much better practice to use either expansion bolts, or else locate, and place adjustable sockets, and inserts in the concrete work.

Often the conduits for wires are placed on the centering and imbedded in the concrete slabs. If the pipes are not too large, and are located with due re-

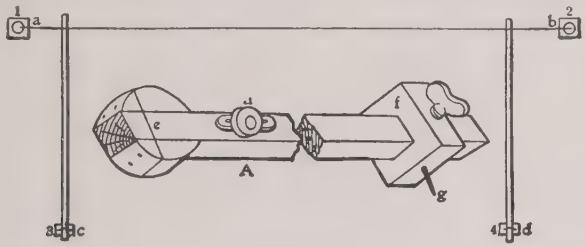


FIG. 89.
STATION ROD.

gard to the reinforcing metal, there is no objection to this method. Piping for sprinklers is generally planned before the concrete work is started. It is therefore a simple matter to place inserts, or sockets where needed in the concrete work, to which the pipes can be readily attached later on.

Machinery Supports.—Large Machines with heavy bases setting on the floor, seldom require any special fastenings. It is usually sufficient to thoroughly grout with Portland cement, the base of the machine to the concrete floor. Small machines may be fastened directly to the concrete by expansion bolts, or similar

methods, after the floors are concreted, the cost of drilling for these fastenings being but a small item of expense. Sometimes, however, it may be necessary to bolt through the floor in order to hold certain machines rigidly. If through bolts are required, the holes should, if possible, be located before the floor is concreted; otherwise in cutting these holes through the floor, reinforcing bars are liable to be encountered,

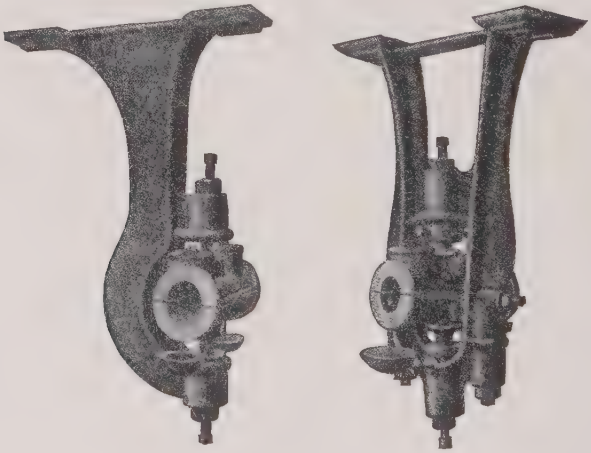


FIG. 90.
MEDART SHAFT HANGERS.

which would cause more or less trouble, and might lead to serious difficulty if cut off. In most cases machinery runs better, and lasts longer if placed on solid foundations, and reinforced concrete floors are admirably adapted for machine shops. There are, however, exceptionable cases where machines seem to require more or less "give" on their bases in order to be most satisfactory in working and endurance. This is particularly true with certain textile machinery, and where a cush-

ion is required a wood finished floor can be laid, or wooden timbers or bolsters can be fastened to the concrete flooring. It has also been suggested to use sheet cork or linoleum carpet under the base of machines in order to give elasticity of movement and also lessen the noise of running machinery.

Lining Out, and Leveling for Shafting.—When the bridge-trees have been placed according to the drawing, a line should be stretched either above, or below the timbers, beams or piers, that are to form the support for the shaft, and parallel with the line to which the work is to be done. If when surveying for the foundations of the buildings, permanents stations have been established, it will be an easy matter to work from these stations and transfer a line from them to the bridge-trees, using a station rod, Fig. 89, in connection with the transit.

If two station stones have been located directly under the center of the shaft, proceed in the following manner: Set up the transit over one stone, and from this point pick up the center mark on the other station stone. Now have an assistant place a pencil on the edge of one of the bridge-trees, then sight to the bridge-tree with the telescope, directing the movement of the pencil in one or another until it is cut by the cross-hairs of the transit, then have the assistant mark the spot thus located and one point in the shaft line will be located. Proceed in this manner to mark each bridge-tree, after which the pillow blocks may be located directly to the center marks which have been made. Some workmen prefer to stretch a line from which to work, instead of working directly to transit-given marks on each bridge-tree. In such cases all that is necessary is to mark the first, and the last timber or pier, and then stretch a line fair with these two marks.

If station stones have not been provided for each shaft, and it is necessary to work from a line some distance at one side, then set up the transit on the line in question, locate one end of the shaft line about where it should come, then set the station-rod, Fig. 89, with the scratch point g upon the shaft center mark on the first bridge-tree. Bring the sliding head of the rod to bear on the cross-hairs, then clamp the rod and carry it to the other end of the shaft line. Here place the rod horizontally as before and at right angles to the line of sight

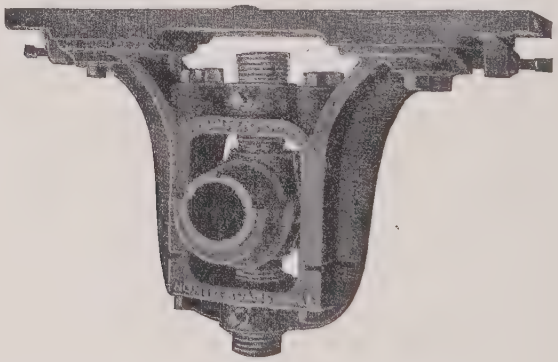


FIG. 91.
HUNTER HEAD SHAFT HANGER.

(this must also be done at the first station), and move the rod until the head intercepts the line of sight. While the rod is in this position, make a mark on the bridge-tree with the scratch point in the rod, and this mark will be in the desired shaft line.

The points thus found may be carried up or down by means of a plumb-line or a spirit-level. A good way to get the marks to the desired vertical height is to nail a bit of board in a vertical position upon the marks on the bridge-trees. Then a line may be stretched at the de-

sired height. The line thus stretched may be used as a center from which to lay out on either side the distance the bridge-tree is to be boxed down to receive the pillow-blocks.

Having established a center line of direction for the shaft, the next move is to level from one bridge-tree to the next and so on, until each one is accurately marked

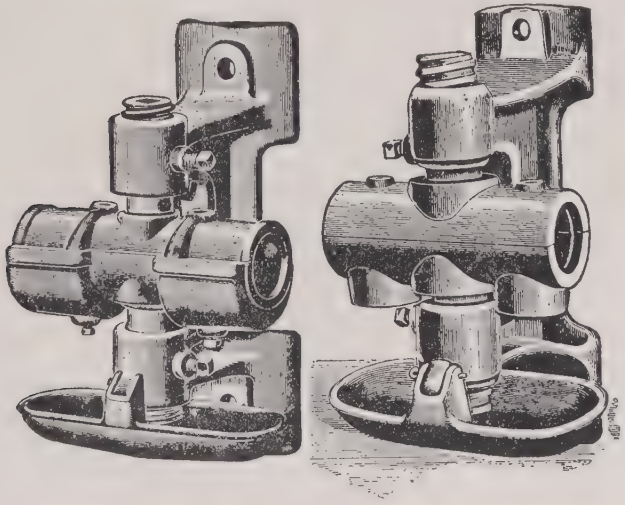


FIG. 92.
GOLDEN'S POST HANGERS.

for the depth of cut required for the pillow blocks. This may be done with the straight-edge and carpenter's level or with the telescope level. A sight may be taken across, either above or below the bridge-trees and measurements taken from this line to the cutting level on each bridge-tree. The preceding remarks apply mainly to cases where the shaft is to be supported upon timber beams, posts, or girders. The same methods are to be pursued in lining out, or leveling for shafting supported

upon concrete or steel construction, with the exception that the material to be dealt with is different.

Various methods of attaching shaft hangers, and brackets to reinforced concrete construction have already been described and illustrated. In cases where it is necessary to put up shafting on old concrete work, either in making changes, or in fitting up buildings not originally intended for mill work, the hangers must be

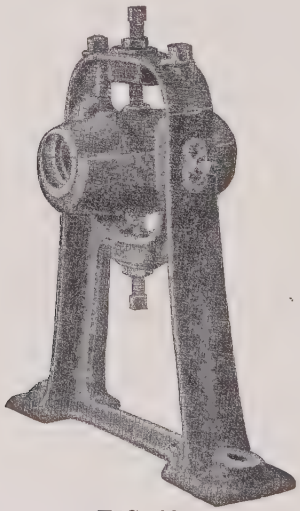


FIG. 93.
HUNTER FLOOR STAND.

bolted to the concrete beams, necessitating the drilling of numerous holes in the concrete.

Shaft Hangers and Floor Stands.—Figs. 90 and 91 show designs for supporting shafting from the ceiling. Fig. 92 is an example of those suited to posts and columns, and Fig. 93, illustrates the Hunter floor stand. The difference in hangers will be found mostly in the methods of adjustment of the boxes, and the facilities

for lubrication. Any hanger that may be selected should have the proper distribution of metal to withstand any strain that may in ordinary service be brought to bear, without danger of breaking or springing. There must also be suitable adjustment in all directions for aligning the shaft when it is being placed in position. The distance between hangers will of course vary with the size of the shafting, and the work that it is expected to do, as for instance the number of pulleys to be carried, and the amount of power that is to be transmitted by it, the arrangement of the machines to which the power is to be transmitted, and the speed of the shaft.

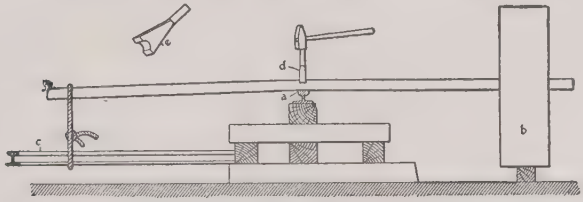


FIG. 94.

All shafting must be true to size and straight if good results are to be obtained, and in order that pulleys may be properly fitted, and when in place, will run true and accurately. The satisfaction that will be obtained from the use of shafting depends, to a large extent, on the arrangement of the machines which are driven from it. As a rule, shafting requires to be straightened after the key seats have been cut, and the couplings have been fitted on.

Fig. 94 illustrates one method that may be used in the absence of a wheel press. The bend is placed on the steel rail, or other hard bearing, with the hollow side of the shaft uppermost, although it appears in the cut to be bent downwards due to the fact that one end has been

loaded with the pulley b, and other weights, while the other end is weighted with an old steel rail c. Enough weight should be placed to bring the ends of the shaft about three inches below a horizontal line throught the bearing a, and the ends of the shaft when not loaded. With this temporary strain on the shaft, it may be straightened by using a sledge hammer and the tool e, applied as shown at d. The end of tool e should be made to fit the circle of the shaft, in order to prevent the shaft being flatened or otherwise distorted by the blows from the sledge hammer. In this manner the "short" or hollow side of the shaft may be elongated, but in order to determine just how much of this treatment is necessary, the shaft should frequently be unloaded and revolved between centers, or other supports at the ends. By repeated and careful work in peening, the shaft may be accurately straightened by this method.

Setting Up and Babbitting Journal Bearings.—While on the subject of shaft erection, a space will be devoted to the above very important matter. First, the pillow blocks should have solid bearings and rest firmly on their support. If in cutting in, the mistake has been made of cutting too deep for the bearing, the best remedy is to make the cut deep enough to allow of the placing of piece of board of standard thickness ½, ¾ or ⅞ inches as a base for the pillow block to rest upon. Before bolting it down, the pillow block casting should be rubbed back and forth on its bearing to ascertain whether or not there are any high places on the surfaces of the bearing, or the face of the casting. The bearing should be rubbed with blue or red chalk which will, when the two surfaces are brought together, clearly indicate the high places.

TABLE 19—U. S. STANDARD BOLTS AND NUTS.

DIA. OF SCW.	THDS PER INCH	DIA AT ROOT OF THD	AREA IN SQ. IN AT ROOT OF THD	HEX. HEAD BOLTS					HEX. NUTS			SQUARE HEAD BOLTS					SQUARE NUTS			TAP DRILL SIZE	TENSION AT 10,000 LB. PER SQ. INCH.	SHEAR AT 8,000 LB. PER SQ. INCH.
				FIN. HEAD		RGH. HEAD		THICKNESS OF HD.	FINISHED			FIN. HEAD		RGH. HEAD		THICKNESS OF HD.	FINISHED					
				SHORT DIA. OF HEAD	LONG DIA. OF HEAD	SHORT DIA. OF HEAD	LONG DIA. OF HEAD		SHORT DIA.	LONG DIA.	THICKNESS	SHORT DIA. OF HEAD	LONG DIA. OF HEAD	SHORT DIA. OF HEAD	LONG DIA. OF HEAD		SHORT DIA.	LONG DIA.	THICKNESS			
1/4	20	.185	.027	7/16	15/32	1/2	9/16	1/4	7/16	15/32	3/16	7/16	5/8	1/2	23/32	1/4	7/16	5/8	3/16	3/16	269	392
5/16	18	.240	.045	17/32	19/32	19/32	11/16	19/64	17/32	19/32	1/4	17/32	3/4	19/32	27/32	19/64	17/32	3/4	1/4	1/4	454	615
3/8	16	.294	.068	5/8	11/16	11/16	25/32	11/32	5/8	11/16	5/16	5/8	7/8	11/16	31/32	11/32	5/8	7/8	5/16	5/16	679	884
1/2	13	.400	.126	13/16	29/32	7/8	1	7/16	13/16	29/32	7/16	13/16	1 5/32	7/8	1 1/4	7/16	13/16	1 5/32	7/16	27/64	1257	1571
5/8	11	.507	.202	1	1 1/8	1 1/16	1 7/32	17/32	1	1 1/8	9/16	1	1 13/32	1 1/16	1 1/2	17/32	1	1 13/32	9/16	17/32	2019	2454
3/4	10	.620	.302	1 3/16	1 11/32	1 1/4	1 7/16	5/8	1 3/16	1 11/32	11/16	1 3/16	1 21/32	1 1/4	1 3/4	5/8	1 3/16	1 21/32	11/16	41/64	3019	3534
7/8	9	.731	.420	1 3/8	1 9/16	1 7/16	1 21/32	23/32	1 3/8	1 9/16	13/16	1 3/8	1 15/16	1 7/16	2 1/32	23/32	1 3/8	1 15/16	13/16	3/4	4197	4810
1	8	.837	.550	1 9/16	1 25/32	1 5/8	1 7/8	13/16	1 9/16	1 25/32	15/16	1 9/16	2 3/16	1 5/8	2 9/32	13/16	1 9/16	2 3/16	15/16	55/64	5502	6283
1 1/8	7	.940	.694	1 3/4	2	1 13/16	2 3/32	29/32	1 3/4	2	1 1/16	1 3/4	2 15/32	1 13/16	2 9/16	29/32	1 3/4	2 15/32	1 1/16	31/32	6940	7952
1 1/4	7	1.065	.893	1 15/16	2 7/32	2	2 5/16	1	1 15/16	2 7/32	1 3/16	1 15/16	2 23/32	2	2 13/16	1	1 15/16	2 23/32	1 3/16	1 3/32	8908	9817
1 3/8	6	1.160	1.057	2 1/8	2 7/16	2 3/16	2 17/32	1 3/32	2 1/8	2 7/16	1 5/16	2 1/8	3	2 3/16	3 3/32	1 3/32	2 1/8	3	1 5/16	1 1/64	10541	11872
1 1/2	6	1.284	1.295	2 5/16	2 21/32	2 3/8	2 3/4	1 3/16	2 5/16	2 21/32	1 7/16	2 5/16	3 3/32	2 3/8	3 3/8	1 3/16	2 5/16	3 3/32	1 7/16	1 5/16	12948	14136
1 5/8	5 1/2	1.389	1.515	2 1/2	2 7/8	2 9/16	2 31/32	1 9/32	2 1/2	2 7/8	1 9/16	2 1/2	3 17/32	2 9/16	3 5/8	1 9/32	2 1/2	3 17/32	1 9/16	1 25/64	15149	16524
1 3/4	5	1.491	1.744	2 11/16	3 3/32	2 3/4	3 3/16	1 3/8	2 11/16	3 3/32	1 11/16	2 11/16	3 25/32	2 3/4	3 7/8	1 3/8	2 11/16	3 25/32	1 11/16	1 17/32	17460	19242
1 7/8	5	1.615	2.048	2 7/8	3 5/16	2 15/16	3 13/32	1 15/32	2 7/8	3 5/16	1 13/16	2 7/8	4 1/32	2 15/16	4 1/8	1 15/32	2 7/8	4 1/32	1 13/16	1 5/8	20490	22080
2	4 1/2	1.712	2.302	3 1/16	3 17/32	3 1/8	3 5/8	1 9/16	3 1/16	3 17/32	1 15/16	3 1/16	4 11/32	3 1/8	4 7/16	1 9/16	3 1/16	4 11/32	1 15/16	1 3/4	23020	25133
2 1/4	4 1/2	1.962	3.023	3 7/16	3 31/32	3 1/2	4 1/16	1 3/4	3 7/16	3 31/32	2 3/16	3 7/16	4 27/32	3 1/2	4 15/16	1 3/4	3 7/16	4 27/32	2 3/16	2	30234	31809
2 1/2	4	2.176	3.715	3 13/16	4 13/32	3 7/8	4 1/2	1 15/16	3 13/16	4 13/32	2 7/16	3 13/16	5 13/32	3 7/8	5 1/2	1 15/16	3 13/16	5 13/32	2 7/16	2 7/32	37189	39270

In boring or drilling the bolt holes, care should be exercised that they be bored fair with the holes in the casting.

When preparing journal bearings, for babbitting, two very important points should not be overlooked, viz: to see that they are clean, and dry. All dirt, and dust should be thoroughly cleaned from the cavities in the castings, after which the casting should be dried by being placed over a forge fire, or if too heavy to be handled in this way, it may be dried by pouring a small quantity of gasoline into the spaces to be babbitted and then setting the fluid on fire. This is in fact a better plan than drying over a fire for the reason that the gasoline will find its way into all the holes and small cavities, thus insuring a thorough drying. A very small quantity of water left in the space to be babbitted will cause lots of trouble when the hot metal is poured in. If the surfaces which are to receive the babbitt metal are heated to a temperature slightly lower than the melting point of the soft metal, the conditions for pouring the bearing may be considered as ideal.

Table 20 gives the constituents of most of the prominent bearing metals as analyzed at the Pennsylvania Railroad laboratory at Altoona, Penn.

The practice of lining journal-boxes with a metal that is sufficiently fusible to be melted in a common ladle is not always so much for the purpose of securing anti-friction properties as for the convenience and cheapness of forming a perfect bearing in line with the shaft without the necessity of boring them. Boxes that are bored, no matter how accurate, require great care in fitting and attaching them to the frame or other parts of a machine.

TABLE 20.

BEARING-METAL ALLOYS.

Trade Name of Alloy	Copper	Tin	Lead	Zinc	Antimo'y	Iron.
Camelia metal	70 20	4 22	14 75	10 20		0 55
Anti-friction metal	1.60	98 13				trace
White metal			87 92		12 08	
Car-brass lining		trace	84 87		15 10	
Salgee anti-friction	4 01	9 91	1 15	85 57		
[1] Graphite bearing-metal		14 38	67 73		16 73	
Antimonial lead			80 69		18 83	
[2] Carbon bronze	75 47	9 72	14 57			
[3] Cornish bronze	77 83	9 60	12 40			
Delta metal	92 39	2 37	5 10			0 07
[4] Magnolia metal			83 55	trace	16 45	trace
American Anti-friction metal			77 44	0 98	19 60	0.65
Tobin bronze	59 00	2 16	0 31	38.40		0.11
Graney bronze	75 80	9 20	15 06			
Damascas bronze	76 41	10 60	12 52			
[5] Manganese bronze	90 52	9 58				
[6] Ajax metal	81 24	10 98	7 27			
Anti-friction metal			88 32		11 93	
Harrington bronze	55 73	0 97		42 76		0.68
Car-box metal			84 33	trace	6 03	
[7] Phosphor bronze	79 17	10 22	10 61			
[8] P.R.R. "B" metal	76 80	8	15 00			
[9] Babbitt metal (parts)	1	50			5	
Babbitt metal light	1 8	89 3			8 9	
Babbitt metal best	3 7	88 9			7 4	
[10] Babbitt (parts) metal best	4	96			8.	
Another "babbitt"	1 5	45 5	40 00		16	
Brittania	1	85 7		2 9		
Brittania		81 9		1 9	16.2	
Brittania	2	81		1	16	
Brittania	4	70 5			25.5	
Brittania	10	22		6	62	
[11] Plate pewter	1 8	89 3			7 1	
White metal bearings on German locomotives	5	85			10	
French white metal h'd	10		65		25	
French white medium	5 5	83 3			11 2	
French white soft		10	70		20.	
French white very soft		12	80		8.	
English—"Parsons"	2	86	2	27	1	
English —"Richards"	4 5	70	10 5		15.	
English —"Babbitt"	3 5	55	23 5		18	
English —"Fenton's"	5	16		79		
English —"French Navy"	7	7 5	7	87 5		
English—"German Navy"	7 5	85			7 5	
Type metal, soft, to			83		17	
Type metal, hard			80		20	
Ornamental castings			66 66		33 33	
[12] Pattern metal		10	10		2	

[1] Contains no Graphite. [2] Contains a possible trace of carbon. [3] Trace of zinc, Iron and Phosphorus. [4] Dr. H. C. Torrey says this analysis is erroneous and that Magnolia metal always contains tin. [5] Contains no Manganese. [6] Phosphorus or Arsenic 0.37. [7] Phosphorous 0.94. [8] Phosphorous 0.20. [9] (In parts by weight). [10] (In parts by weight). [11] Bismuth 1 8. [12] Bismuth 6. Brass 8.

It is not good practice, however, to use the shaft for the purpose of casting the bearings, especially if the shaft be steel, for the reason that the hot metal is apt to spring it; the better plan is to use a mandrel of the same size or a trifle larger for this purpose. For slow-running journals, where the load is moderate, almost any metal that may be conveniently melted, and will run free will answer the purpose. For wearing properties, with a moderate speed, there is probably nothing superior to pure zinc, but when not combined with some other metal it shrinks so much in cooling that it cannot be held firmly in the recess, and soon works loose; and it lacks those anti-friction properties which are necessary in order to stand high speed.

For line-shafting, and all work where the speed is not over 300 or 400 R. P. M., an alloy of 8 parts zinc and 2 parts block-tin will not only wear longer than any composition of this class, but will successfully resist the force of a heavy load. The tin counteracts the shrinkage, so that the metal, if not overheated, will firmly adhere to the box until it is worn out. But this mixture does not possess sufficient anti-friction properties to warrant its use in fast-running journals.

Among all the soft metals in use, there are none that possess greater anti-friction properties than pure lead; but lead alone is impracticable, for it is so soft that it cannot be retained in the recess. But when by any process lead can be sufficiently hardened to be retained in the boxes without materially injuring its anti-friction properties, there is no metal that will wear longer in light, fast-running journals. With most of the best and most popular anti-friction metals in use and sold under the name of Babbitt metal, the basis is lead.

Lead and antimony have the property of combining

with each other in all proportions without impairing the anti-friction properties of either. The antimony hardens the lead, and when mixed in the proportion of 80 parts lead by weight with 20 parts antimony, no other known composition of metals possesses greater anti-friction or wearing properties, or will stand a higher speed without heat or abrasion. It runs free in its melted state, has no shrinkage, and is better adapted to light high-speeded machinery than any other known metal. Care, however, should be exercised in using it, and it should never be heated beyond a temperature that will scorch a dry pine stick.

Many different compositions are sold under the name of Babbitt metal. Some are good, but more are worthless; while but very little genuine Babbitt metal is sold that is made strictly according to the original formula. Most of the metals sold under that name are the refuse of type-foundries and other smelting-works, melted and cast into fancy ingots with special brands, and sold under the name of Babbitt metal.

It is difficult at the present time to determine the exact formulas used by the original Babbitt, the inventor of the recessed box, as a number of different formulas are given for that composition. Tin, copper, and antimony were the ingredients, and from the best sources of information the original proportions were as follows:

		Another writer gives:
50 parts tin	= 89.3%	83.3%
2 parts copper	= 3.6%	8.3%
4 parts antimony	= 7.1%	8.3%

The copper was first melted, and the antimony added first and then about ten or fifteen pounds of tin, the whole kept at a dull-red heat and constantly stirred

until the metals were thoroughly incorporated, after which the balance of the tin was added, and after being thoroughly stirred again, it was then cast into ingots. When the copper is thoroughly melted, and before the antimony is added, a handful of powdered charcoal should be thrown into the crucible to form a flux, in order to exclude the air and prevent the antimony from vaporizing; otherwise much of it will escape in the form of a vapor and consequently be wasted. This metal, when carefully prepared, is probably one of the best metals in use for lining boxes that are subjected to a heavy weight and wear; but for light fast-running journals, the copper renders it more susceptible to friction,

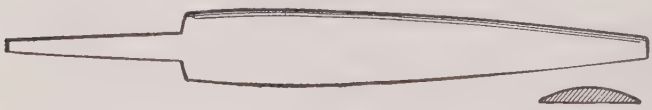


FIG. 95.

and it is more liable to heat than the metal composed of lead and antimony in the proportions just given.

Preparations for Pouring Babbitt Metal.—Provide two pieces of shafting for mandrels of a length somewhat longer than the bearing or box to be babbitted. One of these mandrels is to be turned to a size about $\frac{1}{32}$ inch larger than the diameter of the shaft, and this is to be used for shaping the babbitt in pouring.

The other mandrel should be of the same diameter as the shaft which is to run in the bearing, and this mandrel is for use when scraping the babbitt metal to a bearing.

If it is inconvenient to procure a piece of shafting larger in diameter than the journal, the difficulty may be gotten over by placing a sheet of strong smooth paper

around the mandrel that is of the same diameter as the shaft and then pour the metal around it.

The thickness of the paper around the mandrel, and the contraction of the soft metal when cooled will serve to relieve the journal from being pinched by the babbitt metal, which is liable to occur when the metal is poured directly around the journal. Two thin pieces of wood liners, one for each side of the box, should also be prepared, and they should be of about the thickness of the permanent liners to be used. One edge of these wooden liners is to be notched so as to allow the metal when poured from above to fill the lower box.

These liners are placed between the cap and bottom half of the box with the shaft in a central position and the parts of the box clamped together. In case the box is secured together by means of bolts the outer edges of the liners mentioned are slotted to fit these bolts. Small wooden blocks, cut taper, are placed in each end of the box to maintain the shaft in a central position. The box is placed horizontally in a vise and the shaft centered by means of calipers, the shaft being moved by moving the wedges in or out as needed. After the shaft is correctly located, the ends of the box are stopped by means of putty, or any other convenient method which will prevent the hot babbitt from flowing out. A putty funnel is also built up about the hole into which the hot babbitt is to be poured.

The babbitt is heated in any convenient way until it is fluid. Care should be taken, however, that it does not get too hot, for if it does its efficiency will be diminished. A good method of testing is to dip a piece of soft pine into the hot metal, and when it is just hot enough to char the wood the proper temperature has been reached. It is then poured into the box through the funnel already

prepared, as rapidly as possible, as it quickly sets and if this condition occurs the box will be spoiled and must be cleaned and repoured.

After the metal has set the sprue in the funnel is broken from the cap, the cap removed from the lower part of the box and the shafting removed. The babbitt should have formed a reasonably smooth surface, and be free from cold shots. In cold weather it is sometimes advisable to warm the shaft used as a mandrel.

When the cap and the box are poured separately, the melted babbitt metal is usually poured into one side of the bearing and rises along the other side until the cavity is full.

When pouring a bearing of this character, move the ladle from one end of the bearing to the other, causing the stream to traverse along the slot, thereby keeping the babbitt at approximately the same temperature in all parts of the bearing. When two ladles are used to pour from, it will be sufficient to pour into diagonally opposite corners of the bearing, and if both streams be poured quickly, the cavity will be filled before the metal becomes too cold to flow to the corners.

Oil-channels may be formed in the lining by means of a small, hard-spun cord wound around the mandrel in the direction that the shaft is to run. The cord should be rubbed smooth with putty before being placed upon the mandrel.

Pouring Thin Solid Boxes.—It was stated above that the temperature of the alloy when poured should be just high enough to slightly char a soft-wood stick. This is a general rule, and by following it the millwright will not go far astray on ordinary work. But there are exceptions: when thin solid boxes must be poured, and there is a long narrow or thin space to be filled with soft

metal, then it is sometimes necessary to pour the metal hotter than described.

In such cases, the metals to come in contact with the soft metal should always be heated as much as possible before pouring in the lining.

When a ladleful of metal fails to fill a box, and another ladleful is not at hand, do not try to fill the box by pouring hot metal on top of the cold, for it is very seldom that such a course results in a satisfactory bearing. Hot babbitt will not unite with or weld itself to cold babbitt by simply pouring one on top of the other; therefore, should a man fail to fill a bearing when pouring in the lining, it is decidedly the best practice to take

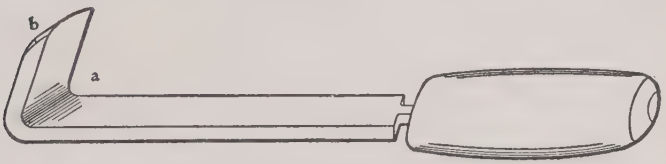


FIG. 96.

down and chip out the partially filled bearing, set it up again, and pour from a supply sufficient to fill the bearing.

Pouring Cold Boxes.—When babbitt or other lining metal is poured into a cold casting, the lining becomes chilled before the iron casting becomes heated; therefore when the soft lining finally cools it shrinks away from the casting and becomes loose, and must be peened to make it tight in the journal bearing casting. Heating the box before pouring in the soft metal lining will prevent this looseness, as the iron casting then shrinks with the soft metal lining on cooling and holds it fast when cold.

There is another way of fastening a lining tightly into

a cold casting and that is by using one of the antimony alloys as a lining. Antimony, when alloyed with lead and with some other soft metals, loses the power of shrinking during the freezing process, and like water, expands and fills the casting so tightly that there is no rattle or looseness. Therefore, when forced to line a cold bearing, use an antimony alloy. It is for this purpose that the antimony alloy is used for casting printing type. The expansion of the alloy during the instant of solidifying or freezing causes the metal to expand into every corner of the mold, thereby securing the extreme sharpness necessary in printing type. The same is true with the antimony alloy in the journal bearing.

Scraping a Bearing.—In order to obtain a good and reliable journal bearing the babbitt should, after cooling, be scraped and accurately fitted to the journal that is to run in it, and for this work the mandrel previously referred to which has been turned to the same size as the shaft, is now covered with ultramarine, and placed in the lower half of the box, and revolved. Upon removing it the high spots in the babbitt will be plainly marked.

There may be only two or three spots so marked at the first trial, but these indicate that the shaft touches the bearing only at those spots, and the metal should be removed at those points by scraping, after which the mandrel should again be placed in the bearing and revolved. This time the shaft may touch in four or five places, as indicated by the markings. This process should be repeated until the bearing spots appear practically all over the surface of the babbitted box, when it may be assumed that the job is complete.

Plumbago, or red lead mixed with oil may be used on the mandrel for the marking in the absence of ultra-

marine, and the mandrel should be rubbed with it each time it is tried in the bearing. The scraping is done with a tool, the usual form of which is shown in Fig. 95.

This tool may be forged from any convenient bit of tool steel, or, as in the case of the tool illustrated, it may be made from a worn-out half-round file. The length of the file should be somewhat longer than that of the bearing to be scraped in order that the tool may be held easily by the hands, one end of the tool projecting past one end of the box while in use.

To scrape a box, grasp the tool with the thumb and fingers of each hand, bring the sharp edge flat against the box lining and shave off the projecting lumps and high spots. For small bearings, the scraper may be used with the round side against the soft metal lining; but when a bearing of large diameter is to be scraped, the flat side of the scraper may be laid against the work. For rapid work, the tool shown in Fig. 96 may be used to advantage. This tool is also made from an old file, but in this case the end of the file, not its side, is made to do the work.

This tool is also a home-made affair, and almost any old flat file may be utilized for making it—something of an advantage when it is considered that the half-round file is seldom found around a plant, while the flat file is to be had for the picking up. But there is quite a knack in making the end scraper so that it will do fast work. There are two things which make this scraper either a good tool or an exceedingly bad one. These things are the angle at which the end of the tool is turned, and the level to which the cutting edge is ground. These things the millwright must experiment with and determine them for the work in hand.

The angle of the bend at a, Fig. 96, is approximately

75 degrees, and the bevel b is about 35 degrees with the bent-up portion, or about 40 degrees from the body of the tool. In using this scraper, the handle is usually lifted more or less out of parallel with the line of cut, which is, of course, lengthwise of the bearing; hence the angle of 40 degrees may be reduced as much as is found necessary to keep the tool cutting smoothly. Should there be any indication of chatter, it is evidence that the tool is not held at the proper angle, which should be changed until the scraper cuts smoothly. As merely

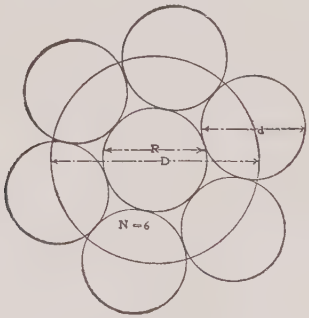


FIG. 97.
SIX BALL BEARING.

raising or lowering the hand changes the angle of cut, the change may be made instantly and without the necessity for grinding or honing. Every scraper should be kept "razor sharp" when scraping is to be done, for good work cannot be done easily with dull scrapers.

Ball- and Roller-Bearings.—The great decrease in friction of shaft and other journals by fitting them with ball- or roller-bearings is not only due to the replacing of sliding, by rolling friction, as many people suppose, for well lubricated journal-bearings have a coefficient of friction of 0.09 to 0.15, while roller-bearings show

coefficients ranging from 0.01 to 0.03. There is another great cause of friction decrease on account of the roller-bearing, and that is: the latter form of bearing is independent to a large extent of lubrication, and the fact of becoming dry does not increase the coefficient of friction as it does with plain journal-bearings where sliding friction must be accounted for.

Ball-bearings are equally efficient with roller-bearings, as friction reducers, but the great trouble with the ball-bearing is that there is not sufficient surface to carry the load, therefore the balls wear out much quicker than the rolls. Unfortunately, increasing the diameter of a ball-bearing does not increase its bearing surface as fast as it increases the inconvenience of using the larger sizes, hence the remedy must be looked for in some other direction.

If a ball-bearing has its length increased to increase the bearing surface, then the ball thus treated becomes a roll, and the transformation into the roller-bearings is complete. Thus a roller-bearing may be considered a ball-bearing with elongated balls. When bearings are mounted upon balls, the aggregate bearing surface presented by even a considerable number of balls is comparatively slight, and a large amount of wear results in a short time, thus limiting the life of a ball-bearing to a very short period. Especially is this the case where heavy journals are in use.

Roller-Bearings.—High grade roller-bearings are made by several concerns, but when bearings of this type are to be attached to a shaft, unless great care is taken that the race, or shell for the rolls is in exact line with the shaft, there will be unequal wear which will more than counteract the saving in power by reduced friction.

The surface of the shaft against which the rollers bear

must be very smooth and true. It must be as perfectly cylindrical as is possible to make it, and the presence of lumps, holes, etc., can only go to add to the coefficient of friction.

Once roller-bearings have been installed, good care should be taken of them. It is a mistake to suppose that a roller-bearing, once in place, will run forever without attention. The roller-bearing must be taken care of. It must be kept clean. As for lubrication, some people claim that lubrication is not necessary for roller-bearings, but any piece of machinery ever built, which will run at all, will run better when well lubricated, and roller-bearings are no exception.

Pin Bearings are a form of roller-bearing used to a considerable extent on push cars in use in factories and mills. These cars are intended to carry heavy loads, and are pushed about by hand on the floor, or on a track as needed. The pin bearings are made of plain short pieces of cold-rolled steel rod, cut to length, and interposed between the journal and housing of the bearing. Three important factors enter in the design of roller and pin bearings, viz., the diameter of the journal, the diameter of the rollers, and the number of rollers. In case the millwright should be called upon to design a roller-bearing, or ball-bearing to fill a given housing upon a shaft whose diameter is known, he may be guided by the following rule and formula: Rule: the diameter of a circle through the center of the balls equals the diameter of a ball divided by the sine of the angle occupied by one-half of the ball. The number of balls and diameter of shaft being known, what shall be the diameter of the balls?

Let N = number of balls,
R = diameter of shaft,
D = diameter of circle through center of balls,
d = diameter of balls,
s = sine.

Assume a journal one inch in diameter is to have a bearing of 6 balls fitted to it

$$\text{then } D = \frac{d}{\text{sine } \frac{180^\circ}{N}}$$

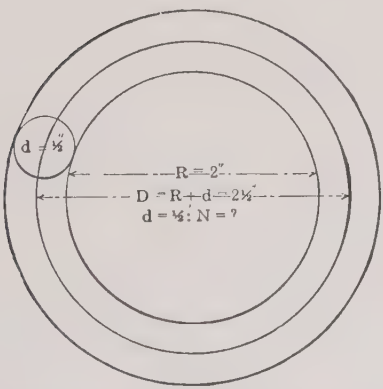


FIG. 98.

in which 180°, half the number of degrees in the entire circle, divided by 6, the number of balls, gives 30° filled by one of the balls in a half circle, which is equivalent to 12 balls in the whole circle, or one-half the angle filled by one ball. By looking in a table of sines in an engineer's reference book, it will be found that the sine of 30 degrees is exactly 0.5, or one-half. The diameter of

the ball being 1 inch, and the diameter of the ball circle being d ÷ 0.5, or ½, it is evident that the ball circle is just twice the diameter of the ball; therefore the balls must be as large as the bearing in order to fill the conditions given as shown by Fig. 97.

Should we have a 2-inch bearing and a 3-inch housing and desire to ascertain the number of pins required to fill it, we may use the equation as follows, the layout being as shown by Fig. 98. As the housing is 1 inch larger than the shaft, the balls must be ½ inch in diameter (not allowing anything for clearance) and the ball circle will be 2½ inches. Hunting up all the numerical quantities that can be found, we obtain:

$$N = ?$$
$$D = 2\tfrac{1}{2} \text{ inches},$$
$$d = \tfrac{1}{2} \text{ inch}.$$
$$R = 2 \text{ inches}.$$

Taking the formula $D = \dfrac{d}{\text{Sine } \dfrac{180^\circ}{N}}$ and substituting the known numbers, we find that $2.5 = \dfrac{.5}{\text{Sine } \dfrac{180^\circ}{N}}$. This reduces to: Sine of $\dfrac{180}{N} = 0.2$. Looking up 0.2 in a table of natural sines, it is found that $0.2 = 11\tfrac{1}{2}$ degrees, nearly. Then $180 \div 11.5 = 15.65$, nearly, therefore 15 pins will go in, and

TABLE 21

POWER THAT STEEL SHAFTING WILL TRANSMIT AT VARIOUS SPEEDS.

From *The Practical Engineer*, September 2, 1892. By A. G. Brown, M.E.

Revs. per Minute.	DIAMETERS OF SHAFTS IN INCHES. HORSE-POWERS THEY WILL TRANSMIT.																			
	1¼	1½	2	2¼	2½	3	3½	4	5	6	7	8	9	10						
50	3·3	5·3	8·0	10·9	15·6	27	43	64	125	216	343	512	729	1000						
60	4·0	6·4	9·6	13·1	18·8	32	51	77	150	259	412	614	875	1200						
70	4·7	7·5	11·2	15·2	21·9	38	60	89	175	302	480	717	1021	1400						
80	5·4	8·5	12·8	17·4	25·0	43	69	102	200	346	549	819	1166	1600						
90	6·0	9·6	14·4	19·6	28·1	49	77	115	225	389	617	922	1312	1800						
100	6·7	10·7	16·0	21·8	31·2	54	86	128	250	432	686	1024	1458	2000						
110	7·4	11·8	17·6	23·9	34·4	59	94	141	275	475	755	1126	1604	2200						
120	8·1	12·9	19·2	26·1	37·5	65	103	154	300	518	823	1229	1750	2400						
130	8·7	13·9	20·8	28·3	40·6	70	111	166	325	562	892	1331	1895	2600						
140	9·4	15·0	22·4	30·5	43·8	76	120	179	350	605	960	1434	2041	2800						
150	10·1	16·1	24·0	32·6	46·9	81	129	192	375	648	1029	1536	2187	3000						
160	10·8	17·1	25·6	34·8	50·0	86	137	205	400	691	1097	1638	2333	3200						
170	11·5	18·2	27·2	37·0	53·1	92	146	218	425	734	1166	1741	2479	3400						
180	12·2	19·3	28·8	39·2	56·3	97	154	230	450	778	1235	1843	2624	3600						
190	12·8	20·4	30·4	41·3	59·4	103	163	243	475	821	1303	1945	2770	3800						
200	13·5	21·4	32·0	43·5	62·5	108	172	256	500	864	1372	2048	2916	4000						
225	15·2	24·1	36·0	49·0	70·3	122	193	288	563	972	1543	2304	3280	4500						
250	16·9	26·8	40·0	54·4	78·1	135	214	320	625	1080	1715	2560	3645	5000						
275	18·6	29·5	44·0	59·8	85·9	149	236	352	688	1188	1886	2816	4009	5500						
300	20·3	32·2	48·0	65·3	93·7	162	257	384	750	1296	2058	3072	4374	6000						
325	21·9	34·8	52·0	70·7	101·6	176	279	416	813	1404	2229	3328	4739	6500						
350	23·6	37·5	56·0	76·2	109·4	189	300	448	875	1512	2401	3584	5103	7000						
375	25·3	40·2	60·0	81·6	117·2	203	322	480	938	1620	2572	3840	5468	7500						
400	27·0	42·9	64·0	87·0	125·0	216	343	512	1000	1728	2744	4096	5832	8000						
425	28·7	45·6	68·0	92·5	132·8	230	364	544	1063	1836	2915	4352	6197	8500						
450	30·4	48·2	72·0	97·9	140·6	243	386	576	1125	1944	3087	4608	6562	9000						
475	32·1	50·9	71·0	103·4	148·4	257	407	603	1188	2052	3258	4864	6926	9500						
500	33·7	53·6	80·0	108·8	156·2	270	429	640	1250	2160	3430	5120	7290	10000						

For power of wrought-iron shafts take 70 per cent. of steel shafts of the same size.

if they are a little smaller than ½-inch in order to give the necessary clearance, 16 pins may go into the housing. In this manner, the millwright can find any of the

quantities pertaining to ball or roller bearings, provided a sufficient number of the points are given.

Set Collars.—Every shaft should be fitted with two set collars, and these should preferably be placed at opposite ends of the same bearing. It is the practice of some to place one set collar at either end of a shaft, but the two collars against the same pillow-block is preferable, as it leaves but one bearing to look to in case the shaft should develop too much end motion. In very long shafts, the variation in temperature has to be taken into account when end collars are used, but with both against the same bearing, and near the middle of the shaft, no attention need be given to temperature variations.

Safety set collars should be used in every case, and no collar with a set-screw projecting beyond the face of the collar should ever be tolerated. Collars are now generally made with a thin flange at either end, the flanges being large enough to project farther than the head of the set-screw. In some cases it is possible to dispense with one set collar, due to the fact that one or more of the pulleys may come next the journal-bearing, thus taking the place of a collar at that point. Countershafts carrying loose pulleys should always have a set collar next the loose pulley opposite the tight pulley, and where a loose pulley comes near a journal box, a collar should be interposed between the box and the hub of the pulley to prevent pinching.

Set collars should never be so placed upon a shaft as to prevent a certain amount of end play of the shaft. This applies especially to line shafting, and all other cases where a slight end motion can be allowed. There are, of course, many machines, such as circular saws and similar machines, where end motion cannot be permitted.

Electric generators and motors will run much better in every respect if they be given from 1/16 to 3/16 inch end motion. The amount of end motion to be given to shafting and machines depends upon conditions and the kind of machinery, but the clearances given above will generally apply.

Pressure Suitable for a Given Velocity of Journal.—When the product of the pressure multiplied by the velocity of a journal exceeds a certain limit there is danger of heating. The working pressure should therefore be considerably less than that at which heating may commence.

The cool-running pressure, and that under which bearings may become heated, also the pressure generally producing very hot bearings, may be determined by the following rule:

Let P = the maximum pressure in pounds per square inch permissible on the horizontal bearing-surface of the journals of a horizontal shaft, with ordinary oiling.

V = the maximum permissible circumferential velocity of the journals of a shaft in feet per minute.

C = A constant = 28,000, for cool bearings.

C = 40,000, for considerably heated bearings.

C = 50,000, for very hot bearings.

The maximum pressure per square inch, P, allowable with ordinary oiling on the horizontal bearing-surface of the journals of a shaft is, $P = C \div V$.

Example—A shaft having journals of 4 inches diameter, makes 100 revolutions per minute. What should be the maximum pressure on the bearings, consistent with freedom from liability to heat?

Then, the circumference of each journal of the shaft

$= 4 \times 3.1416 = 12.5664$ inches $\div 12 = 1.0472$ feet; and 1.0472 feet $\times 100 = 104.72$ feet per minute, the circumferential velocity of each journal of the shaft; and $28{,}000 \div 104.72 = 268$ lbs. per square inch, the maximum pressure allowable on the horizontal bearing-surface of the journals. If the journals are 6 inches long, the horizontal area of bearing-surface $= 4 \times 6 = 24$ square inches; and the bearing capacity of each journal is $= 24 \times 268 = 6{,}432$ lbs.

Velocity of the Journals of a Shaft Suitable for a Given Pressure.—With ordinary oiling, the maximum permissible circumferential velocity in feet per minute of the journals of a horizontal shaft is $V = C \div P$, the notation being the same as in the preceding rule.

Example—A shaft with journals of 5 inches diameter has a pressure of 200 lbs. per square inch on the bearings. What should be the maximum circumferential velocity of the journals consistent with freedom from liability of the bearings to heat?

Then $28{,}000 \div 200 = 140$ feet, the maximum permissible circumferential velocity of the journals per minute. The circumference of each journal of the shaft $= 5 \times 3.1416 = 15.708$ inches $\div 12 = 1.309$ feet, and this shaft may make $140 \div 1.309 = 107$ revolutions per minute.

Length of the Journals and Bearings of Shafts.—The length of the journals of shafts should be in proportion to the weight carried, and to the nature of the bearing-metal. Bearings of white metal, being softer, should be longer than those of bronze, in order to distribute the pressure over a greater surface. The following are good general proportions, in which D = the diameter of the journal in inches.

TABLE 22.—LENGTH OF THE JOURNALS AND BEARINGS OF SHAFTS FOR VARIOUS LOADS AT A SPEED NOT EXCEEDING 160 REVOLUTIONS PER MINUTE.

Description of Bearing-Metal	Length of Journal and Bearings in terms of the Diameter				
	For very light Loads	For light Loads	For ordinary or medium Loads	For heavy Loads	For very heavy Loads
Bronze and gun-metal....	D×1.00	D×1.25	D×1.50	D×1.75	D×2.0
Brass	D×1.25	D×1.50	D×2.00	—	—
Antifriction white-metal...	D×2.00	D×2.50	D×3.00	D×3.50	D×4.0
Cast-iron	D×4.00	D×4.25	D×4.50	—	—

The length of the journals and bearings of the spindles or shafts of emery wheels, fans, and blowers may be $= D \times 5\frac{1}{2} - 7$.

The length of the journals and bearings of the shafts of generators of electricity and other high-speed machines, may generally be as given in the following table:

TABLE 23.—LENGTH OF THE JOURNALS AND BEARINGS OF HIGH-SPEED MACHINES.

Revolutions of the Shaft in Feet per Minute.							
200	250	300	400	500	600	800	1000
D x 2.50	D x 2.75	D x 3.0	D x 3.5	D x 4.0	D x 4.5	D x 5.0	D x 6.0

Bearing Capacity of Journals.—The bearing-surface multiplied by the unit of pressure suitable for the particular bearing gives the bearing capacity of the journal. For instance, the projected area, or horizontal bearing-surface, of a journal of 4 inches diameter, and 6 inches long, is $4 \times 6 = 24$ square inches; and if a pressure of 300 lbs. per square inch be assumed, the bearing capacity of the journal is $24 \times 300 = 7{,}200$ lbs.

The length, L, in inches of the journal of a shaft is found as follows:

$$L = \frac{\text{Bearing capacity of the journal in lbs.}}{\left(\begin{array}{c}\text{Pressure on the bearing in pounds per square inch}\end{array}\right) \times \left(\begin{array}{c}\text{width of the arc of contact of the bearing in inches.}\end{array}\right)}$$

The width, W, in inches of the arc of contact of the bearing is:

$$W = \frac{\text{Bearing capacity of the journal in lbs.}}{\left(\begin{array}{c}\text{Pressure on the bearing in pounds per square inch}\end{array}\right) \times \left(\begin{array}{c}\text{length of the bearing in inches.}\end{array}\right)}$$

The pressure, P, in lbs. per square inch on the projected area of the shaft is:

$$P = \frac{\text{Bearing capacity of the journal in lbs.}}{\left(\begin{array}{c}\text{Length of the bearing in inches}\end{array}\right) \times \left(\begin{array}{c}\text{width of the arc of contact of the bearing in inches.}\end{array}\right)}$$

The usual pressure on various bearings of journals is given in Table 24.

Average Pressure on Various Bearings.—The average lowest, highest, and mean pressures on bearings in practice are generally as given in the following table:

TABLE 24.—PRESSURE ON THE BEARINGS OF THE JOURNALS OF SHAFTS, CRANK-PINS, CROSSHEAD-PINS, AND OTHER BEARINGS, MEASURED PER SQUARE INCH OF THE HORIZONTAL DIAMETRAL SECTION OF THE JOURNALS.

Description	Pressure in Pounds per Square Inch of the Horizontal Sectional Area of the Bearing		
	Lowest.	Highest.	Average.
Bearings of the shafting of factories, with couplings, but without pulleys	12	40	26
Bearings of the shafting of factories, with couplings, and with an average number of pulleys	30	100	65
Bearings of second-motion shafts, and main-shafting of factories, with couplings, pulleys, and toothed-wheel gearing	50	240	145

Final Alignment.—After the pulleys have all been firmly keyed in their several places, the couplings set and properly bolted, and all of the journal bearings have been properly babbitted and the caps of the boxes tightened solidly upon the liners or bottom castings, as the case may be, the next test should be one of final alignment, not only for parallel lines of shafting, but pulley alignment also, in order that the belts when put on may run properly.

In the first place the main line shaft with a length, say of 42 feet and a weight of several tons of pulleys, etc., should, if properly adjusted, revolve so easily that one-man power applied to the rim of the 60-inch main pulley would turn it. If it cannot be so turned, it is an indication of abnormal tightness somewhere along the line, and the bearings should be carefully gone over again and examined. Some of them may need thicker liners. In order to have the belts run true on the pulleys of parallel shafting, it is absolutely necessary that the shafts shall not only lie level in their bearings, but the different lines be exactly parallel with each other, or at least as near so as it is possible to get them. This work may be done with the transit and station rod as already described, and in addition to this a great deal may be accomplished by the following-described method of pulley sighting. Two sights should be taken over every pulley, one sight for each, as shown in Fig. 99, to which reference is now made. Two pulleys, a and b, are to be connected by belt, and should run so true with each other that the belt will show alike on both.. From position c, Fig. 99, the pulleys appear to be exactly in line with each other, and at e the edge of the pulley b coincides perfectly with the entire side of pulley a. But if the eye be directed toward the top of pulley b at f,

a portion of the rim of pulley b comes in sight. At the far side of the pulley there will be more of the rim seen, as at g, showing that while pulley a is in line with e, pulley b is not in line with pulley a, as may be proved by moving the eye to position d, which is in line with the opposite face of pulley b.

When the conditions are met with as in Fig. 99, it is time to do a little investigating and see which shaft is out of line—or which pulley is not true upon its shaft. First, have an assistant revolve pulley (and shaft) a slowly while you continue to sight from e. If no variation in the line of sight is observed at e, while pulley a revolves, it may be considered that pulley a is all right.

FIG. 99.
SIGHTING OVER PULLEYS.

Next, have pulley b revolved slowly, while still looking from a. If no variation is observed at e, it may be assumed that pulley b is also truly fixed upon its shaft.

But should there be observed any variation of side of pulley e as that pulley revolves, then try line of sight d, and have pulley b stopped when the greatest variation is either to the right or to the left, and look at the pulley to see whether it is bored too large for the shaft or if the machinist has omitted facing the sides of the pulley. Occasionally it will be found that the pulley is not bored true with its rim, having in some way moved in the boring mill. In such cases, two courses are open: Either throw away the pulley and put in a perfect one, or place the points of variation vertical, and then sight

across g, e, and move the shaft of pulley b until the sight across g, e coincides with sight from c.

Previous to moving the shaft through b, the alignment of both shafts should be tested, for there is a possibility that the shaft at a, may be the one in error, in which case shaft b should not be moved, and it will be necessary to move one of the pulleys along its shaft in order to make them line up after the shafts have been placed parallel. With pulleys 36 inches in diameter, a great degree of accuracy can be obtained by sighting pulleys as above—accuracy not only in alignment of the pulleys, but provided the pulleys are truly finished and set on their shafts, accuracy in the alignment of the shafts.

Having made all adjustments by sighting the pulleys, a final transit test should be made on all the shafts, or in the absence of the transit, the test may be made with the straight-edge, and spirit level to ascertain whether or not the shafting is level.

To test the alignment of the shaft with the permanent station stones, or targets, place the transit on the station stones and pick up the shaft-line, or take up that line from any targets that may be available for the purpose. The leveling-rod head may be either replaced on its rod, or put upon a piece of plain square rod. Place the free end of the rod against the shaft, hold it horizontal, and set the sliding head to a sight from the instrument cross hairs, then carry the rod to the next bearing, and if the cross hairs cut the same point on the rod, that section of the shaft may be considered all right. Proceed in the same manner to each bearing. Where a section of larger diameter is encountered, the rod must be shortened by moving the head towards the shaft a distance equal to the difference in diameters, and vice versa for a section of shafting of smaller diam-

eter. Any variation found must be corrected by moving the bearing at which the variation is found, an amount sufficient to bring the shaft in line.

For leveling the shaft, the tools shown in Fig. 100 will be found to be very convenient. In using this tool the hook c is hung on the shaft, and as it is made of ½-inch soft steel, it may be placed in a very narrow

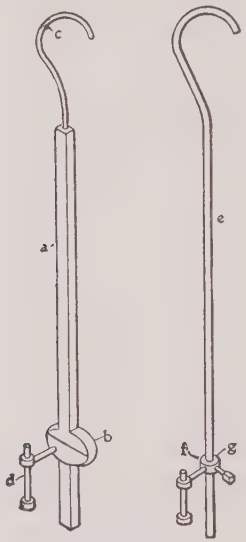


FIG. 100.
LEVELING RODS FOR SHAFTING.

space between pulley and hanger, or hanger and collar. The hook should be so shaped that the rod will hang in a vertical position, thus becoming self-plumbing. A hole is bored in the sliding head b, and a clip of some kind is pushed into the hole for the purpose of supporting the short piece of pipe d, which is capped at its lower end, and partly filled with kerosene into which a wick is inserted, thus forming a small torch which

serves to light up the sliding head in order that a good view of it may be obtained through the telescope.

A good substitute for the leveling rod just described may be made by bending a piece of round steel rod, as shown at e, Fig. 100. The rod may be ½-inch, or any convenient size, and a small set collar, f, answers the purpose of a sliding head, and the torch carrier is attached to this.

Testing Shafts Without the Transit.—When the shaft must be tested without the use of the transit or other telescope instrument, stretch a line conveniently above the shaft and a few inches to one side. Hang a plumb-line opposite the point to be tested, and after the bob-line has come to rest, carefully measure from the line to the shaft, placing a bit of thin stick against the shaft and marking the stick where the bob-line hung past it. Carry the stick to each point in succession which is to be tested.

To test the level of the shaft, a straight-edge should be used and placed upon two distance pieces which will reach from the straight-edge, when placed above the pulleys, down to the shaft. Three men are needed for this work, two to hold the distance pieces and the straight-edge, the other man to work the carpenter's level which is applied to the straight-edge as directed for leveling foundations. The bearings are to be raised or lowered as indicated by the level and straight-edge.

It is a common occurrence to see millwrights apply the level direct to the shaft. This method is all right under certain conditions, but there are times when it is not to be depended upon. Every shaft must sag a little between supports, and if the level be applied near one of the bearings, it will not indicate exactly the same as when applied at the center of a span. Where there are

heavy pulleys a few inches from a bearing, as all pulleys should be placed, and the level is applied just outside of the pulley, there will be found sufficient deflection in the shaft to lead to a considerable error when the shaft is used as a straight-edge.

Proportions of Pulleys.—For the proportions of pulleys the following formulæ are presented by Unwin and Reuleaux: Let n = number of arms, D = diameter of pulley, S = thickness of belt, t = thickness of rim at edge, T = thickness in middle, B = width of rim; B^1 = width of belt, h = breadth of arm at hub, h^1 = breadth of arm at rim, e = thickness of arm at hub, e^1 = thickness of arm at rim, c = amount of crowning in inches; then,

	Unwin	Reuleaux
$B = 9/8\ (B^1 + 0.4)$,		9/8 B^1 to 5/4 B^1
$t = 0.7\ S + .005\ D$,		(thickness of rim)
		1/5 h to 1/4 h

$$T = 2t + C,$$

$$h = \begin{cases} \text{for single belts} = .6337 & \sqrt[3]{\frac{BD}{n}} \\ \text{for double belts} = .798 & \sqrt[3]{\frac{BD}{n}}\ \frac{1/4''}{4} + \frac{B}{4} + \frac{D}{20\,n} \end{cases}$$

$h^1 = 2/3\ h$		0.8 h
$e = 0.4\ h$		0.5 h
$e^1 = 0.4\ h$		0.5 h

$$n = \begin{cases} \text{number of arms} & 3 + \frac{BD}{150} \quad \text{to} \quad \frac{1}{2}\left(5 \times \frac{D}{2\,B}\right) \\ \text{for a single set} & \end{cases}$$

$$\text{length of hub} = \begin{cases} \text{not less than } 2.5\ S \\ \text{is often } 2/3\ B \end{cases} \Bigg\} \begin{matrix} B \text{ for single arm pulleys} \\ 2\,B \text{ for double arm pulleys} \end{matrix}$$

M = thickness of metal in hub, h to ¾ h
c = crowning of pulley 1/24 B

According to Unwin, the number of arms is arbitrary, and may be altered, if necessary. Reuleaux says, that pulleys with two or three sets of arms may be considered as two or three separate pulleys combined in one, except that the proportions of the arms should be 0.8 or 0.7 times that of single-arm pulleys.

Example—Dimensions of pulley as follows: Diameter 60″, face 16″, for double belt ½″ thick;

Solution by

	n	h	h^1	e	e^1	t	T	L	M	c
Unwin	...9	3.79	2.53	1.52	1.01	.65	1.97	10.7	3.8	.67
Solution by										
Reuleaux	..4	5.0	4.0	2.5	2.0		1.25		16	5

The following solution for breadth of arm is proposed by Prof. William Kent: Assume a belt pull of 45 lbs. per inch of width of a single belt, that the whole strain is taken in equal proportions on one-half of the arms, and that the arm is a beam loaded at one end, and fixed at the other. The formula for a beam of elliptical section is $fP = .0982 \frac{Rbd^2}{l}$, in which P = the load, R = the modulus of rupture of the cast iron, b = breadth, d = depth, and l = length of the beam, and f = factor of safety. Assume a modulus of rupture of 36,000 lbs.,

a factor of safety of 10, and an additional allowance for safety in taking $l = \frac{1}{2}$ the diameter of the pulley instead of $\frac{1}{2}D$ less the radius of the hub. Take $d = h$ the breadth of the arm at the hub, $b = e = 0.4\,h$, the thickness. We then have $fP = 10 \times \frac{45\,B}{n \div 2} = 900 \frac{B}{n}$

$$= \frac{3535 \times 0.4\,h^2}{\frac{1}{2}D} \quad \text{whence} \quad h = \sqrt[3]{\frac{900\,BD}{3535\,n}} = \sqrt[3]{\frac{BD}{n}},$$

which is practically the same as the value reached by Unwin from a different set of assumptions.

Convexity of Pulleys.—On the subject of the convexity of pulleys, authorities differ. Morin gives a rise equal to 1/10 of the face; Molesworth, 1/24; others from $\frac{1}{8}$ to 1/9. Scott A. Smith says, the crown should not be over $\frac{1}{8}$-inch for a 24-inch face. Pulleys for shifting belts should be "straight," that is, without crowning.

Cone or Step Pulleys.—To find the diameter for the several steps of a pair of cone pulleys, proceed as follows: First, for crossed belts, let D and d be the diameters of two pulleys connected by a crossed belt, L = the distance between their centers, and B = the angle that either half of the belt makes with a line joining the centers of the pulleys; then total length of belt

$$= (D + d)\frac{n}{2} + (D + d)\frac{n\,B}{180} + 2\,L \cos B,$$

$$\beta = \text{angle whose sine is } \frac{D + d}{2\,L};$$

$$L \cos \beta = \sqrt{L^2 - \left(\frac{D + d}{2}\right)^2} \div L.$$

The length of the belt is constant when D + d is constant; that is, in a pair of step pulleys the belt tension will be uniform when the sum of the diameters of each opposite pair of steps is constant. Crossed belts are seldom used for cone-pulleys, on account of the friction between the rubbing parts of the belt. To design a pair of tapering speed-cones so that the belt may fit equally tight in all positions, first, when the belt is crossed, use a pair of equal and similar cones tapering opposite ways; second, for open or uncrossed belts, use a pair of equal and similar cones tapering opposite ways and bulging in the middle, according to the following formula: Let L = the distance between the axes of the conoids, R = the radius of the larger end of each, r = the radius of the smaller end; then the radius in the middle, r^1, is found as follows:

$$r^1 = \frac{R + r}{2} + \frac{(R - r)^2}{6.28\,L} \text{ (Rankine).}$$

If D^1 = the diameter of equal steps of a pair of cone-pulleys, D and d = the diameters of unequal opposite steps, and L = the distance between the axes, then $D^1 = \frac{D + d}{2} + \frac{(D - d)^2}{12.566\,L}$. If a series of differences of radii of the steps R — r be assumed, then for each pair of steps $\frac{R + r}{2} = r^1 - \frac{(R - r)^2}{6.28\,L}$, and the radii

of each may be computed from their half sum, and half difference, as follows:

$$R = \frac{R+r}{2} + \frac{R-r}{2}; \qquad r = \frac{R+r}{2} - \frac{R-r}{2}.$$

While it is good practice to run shafting at a high rate of speed, care must be exercised in order not to make it impossible to install the proper pulleys for driving the machines, at the correct speed. The speed of the shafting bears a direct relation to the size of pulleys required to obtain a certain increase or decrease in speed, and cases have been known in which the size of pulley necessary was so large that it could not be installed. This condition makes it imperative to obtain the correct speed by means of several reductions, and is not only a waste of power, but calls for extra belting, apparatus, lubricant and care.

Lagging Pulleys.—Changes in a mill or factory frequently necessitate an increase in the diameter of a pulley; either this or the purchase of a new one of the required diameter but as this would cause considerable expense, the desired change in diameter may be effected by lagging, which may be done quicker and cheaper than the installing of a new pulley. The lagging may be put on so as to increase the diameter 2 inches or even 5 or 6 inches if necessary. This is particularly the case with wooden pulleys, but even cast iron, or steel rim pulleys may be lagged to advantage by drilling a series of holes around the circumference of the rim. A method of laying out a lag pattern is illustrated by Fig. 101. First, procure a thin piece of board ¼ to ½ in. thick and tack it to the floor or bench. Then with a pair of dividers or a tram draw a circle c, from

center b, having a diameter equal to that of the pulley to be lagged. The circle should cut on to the board

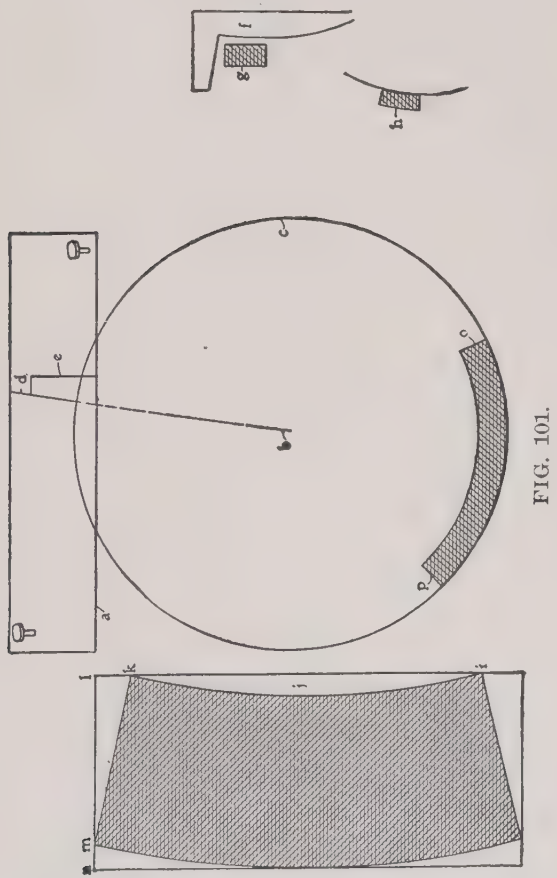


FIG. 101.

about 2½ inches, and a line should then be drawn from center b to d, after which the thin board may be sawed off at e, 2 or 3 inches from line d. The edge of the

board at d, and the end at a, may also be cut off leaving the completed pattern as shown at f.

It is to be used as a set square for testing the dressing of the lags, one of which is shown in section at g, and it is to be planed or machined until it fits the pattern f on three sides. A completed lag is shown at h. The curve i, j, k, may be worked out by running the lags at nearly a right angle across the table of a

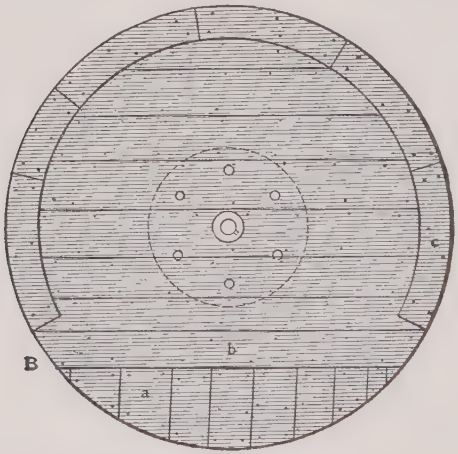


FIG. 102.
CONSTRUCTION OF WOOD PULLEY ON A FLANGE.

large circular saw, having the saw teeth project through the table the depth of the cut at j.

There is no need of dressing down the lag g until pattern f fits all the way across, because at i, k, is shown all that can be used of the curve, the edges k, l, being cut off when the lag is jointed on the edge, the strip k, l, m, being wasted. Hence it is not necessary to fit the distance k, l. The width of the lag across the face i, k, is immaterial, for the reason that the edges are

all radial when laid out by pattern f. Hence any width of stock may be used that will give the necessary thickness of lagging.

Wood Pulleys. — Fig. 102 represents a wood pulley in course of construction on flanges, according to the old time method. Circles or disks of 7/8 or 1 inch dressed lumber are first gotten out to amount to a little less than one-third the width of pulley face. These circles are fitted to each other, glued and nailed as shown at a and b, sketch B. When the requisite thickness of web, as it is called, has been built up, the flange is bolted to the wooden web, and usually another thickness of stuff is broken around the outside of the flange, thus giving more bearing of the wood against the iron.

As soon as the web has been completed, the segments c are fitted in place, nailed and glued, until the necessary width of face has been worked up. After the nails have all been carefully "set," the pulley is mounted on a shaft, and the face, and edges of the face are turned down sufficiently to obtain the needed finish. When gluing the disks and segments together, the pieces should be fitted to lie as closely as possible, the coating of glue applied, and the whole squeezed in a press as tight as possible, while the glue it setting.

To prepare hot glue for use, place the required quantity of dry glue in a vessel and cover with cold water. It makes no difference how great a quantity of water is used for the glue will only absorb a certain amount. It is best to place the glue in the water at night; then the next morning pour off all the water except what has been absorbed by the glue, which should then be placed in the regular double glue pot and melted. Dry glue will absorb just the amount of cold water that is necessary for melting the glue with heat.

Glue will not dissolve in cold water, though it will soften, as shown above. To prepare a glue which will not turn into jelly after it has been heated and allowed to become cold again, add some acetic acid, and the property of gelatizing when cold becomes lost. Nitric acid may be used instead of acetic, if desired, so may hydrochloric (muriatic) acid. None of the adhesive strength is lost, and glue thus prepared, which is sold as "cold" glue, "prepared" glue, etc., may be used without heating if necessary, but better results can be obtained by heating the glue, likewise the articles to be glued, the same as when using ordinary, or "hot" glue.

Glue joints may be made waterproof by adding a little bichromate of potash to the glue before using, and then exposing the glued and dried joint to the action of sunlight, which acts upon the bichromate and renders the glue insoluble in water.

Belts and Belt Drives.—The use of belting in manufacturing plants, whether of rubber, leather, canvas or rope, brings many problems into the work of the millwright that require much judgment, skill and experience in order to produce satisfactory results at a minimum cost and keep the production factor constant. Any special local conditions in every plant have to be met by a special mode of procedure.

In the selection of a driving belt, the work to be done and the atmospheric conditions surrounding it are of the first importance. Always be sure to so design the drive that the belt will be amply large for the work, but not so large as to cause unnecessary expense in first cost. Figure to have the belt speed in feet per minute between 4,000 and 5,000 feet if possible, as better results may be obtained with smaller belts under these conditions, and this factor of velocity should enter into what-

ever belt formula is used in determining the horsepower, or the width of the belt.

The atmospheric conditions are as variable as the power factor, and the life and efficiency of the driving medium are almost entirely governed by these conditions. In cotton mill work the air is dry and linty, and even with humidifiers working, the belts will occasionally need a liberal amount of neatsfoot oil or other good dressing. Belts for this work are best when made of leather, and what is known as a "soft tan" gives excellent results as regards pliability.

Machine shops are generally dusty and dry as regards atmospheric conditions, and wood working shops, polishing and grinding rooms are invariably so. Even under these conditions if belts are kept soft and pliable, and well laced or cemented they will last almost indefinitely.

In designing belt drives there are several very important features which the millwright should keep constantly before him, and correct data upon these points should be obtained before any figuring is done. For instance he should know:

(a) What is the distance between centers.
(b) The kind of pulleys—iron, wood, paper, etc.
(c) The kind of belt—leather, rubber, cotton, etc.
(d) Is the belt to be single, double or treble?
(e) What velocity can be given the belt?
(f) What horse power is to be transmitted?
(g) Will the resistance to turning be constant?

The tension per inch of belt width that may be allowed is also an important factor. Some designers have adopted 40 lbs. pull per inch of width as a standard for all kinds of belts, leather, rubber and cot-

ton. Others allow 60 lbs., while others still allow 88 lbs., but only for certain kinds of belting. A belt transmits motion entirely through frictional contact with the surface of the pulley.

It is best to have the face of the pulley from ½ inch to 1½ inch wider than the belt, depending upon width of face. Wide belts are less effective per unit of sectional area than narrow belts, and long belts are more effective than short ones. The proportion between the diameters of two pulleys working together should not exceed 6 to 1.

The usual convexity of pulley face, is ⅛ to ¼ inch in the width, up to 12 inches wide, and for larger sizes ⅛ to ¼ in. per foot of width.

Speed of Belts.—Belts have been employed running over 5,800 feet per minute, but nothing is gained by running belts at a speed much in excess of 4,000 feet per minute. About 3,500 feet per minute for main belts is good practice, and for lathe belts from 1,500 to 2,000 feet per minute. Table 25 gives the coefficients of friction of different kinds of belts.

Table 25

Coefficient of Friction.

Leather belt on iron pulley, dry	0.42
Leather belt on iron pulley, wet	0.36
Leather belt on iron pulley, greasy	0.23
Leather belt on leather covered pulley	0.45
Cotton belt on iron pulley, aver	0.30
Rubber belt on iron pulley, aver	0.45

At high velocities the centrifugal tension will decrease the pressure between belt and pulley, besides putting additional tension in the belt.

Rankine's formula for centrifugal tension is

$$C = \frac{12 \times W \times V^2}{g}.$$

in which W = Weight of cu. in. of leather,
C = Centrifugal tension,
V = Velocity in feet per second,
g = Acceleration of gravity = 32.2,
let W = .035,

$$\text{then } C = \frac{12 \times .035 \times V^2}{32.2} = \frac{.42 \times V^2}{32.2} = .013V^2.$$

On high speed belts running at, say over 3,000 feet per minute, the effect of centrifugal tension becomes an important factor. For instance, suppose we had a 6 in. belt ¼ in. thick, with cemented joints; Max. safe load to be 400 lbs. per sq. in. Then $6 \times .25 = 1.5$ sq. in., which $\times 400 = 600$ lbs. If this belt were to be speeded up to 176 feet per second, then C will equal $.013 \times 176^2 \times 1.5 = 604$ lbs., and therefore the centrifugal force would equal the safe tension and the belt could not transmit any power. Thus it is necessary to figure on centrifugal force for all speeds of 3,000 feet per minute or more.

Take another example; a 20-inch belt ¼ inch in thickness is to run at a speed of 2,670 feet per minute, and the pull on the belt is 1,235 lbs., equal to $1,235 \div 20 = 61.8$ lbs. pull per inch width. To this must be added the centrifugal tension. The belt being 20 inches wide by ¼ in. thick, there are $20 \times .25 = 5$ sq. inches of leather in section, then $C = .013 \times 44.5^2 \times$

$5 = 128.7$ lbs. This will increase the total tension on the belt to $1,235 + 128.7 = 1,363.7$ lbs., and the pull per inch width of belt will be $1,363.7 \div 20 = 68.1$ lbs.

Strength of, Working Tension in, and Horse Power Transmitted by, Belts.—The ultimate strength of ordinary bark tanned single leather belting varies from 3,000 to 5,000 lbs. per square inch of cross-section. It is, however, sometimes considered more convenient to state the tenacity of a belt in lbs. per inch of width. The thickness of single belting varies from 3/16 to 5/16 inch, and from 3/8 to 5/8 inch for double belting. Taking the mean thicknesses, we may say that the breaking stresses are from 750 to 1,250 lbs. per inch of width for single belts, and 1,500 to 2,500 lbs. per inch of width for double belts.

It is found, however, that the strength of the joints is sometimes only about one-third of the strength of the solid leather. Hence, the *final* strength of an endless belt should not be reckoned at more than *one-third* of the above values. Further, the *safe working tension* should never exceed *one-fifth* of this final strength (of the joint), in order to provide for deterioration and sudden changes of load. From this point of view, we thus arrive at approximate values for the safe working tensions by taking one-fifteenth (*i.e.*, one-third for joint and one-fifth for factor of safety) of the breaking stress *per inch of width*, viz.:

For single belting, 50 to 80 lbs.
For double belting, 100 to 160 lbs.

In *practice*, however, the rule is not to put on a greater tension than 50 lbs. for single, and 80 lbs. for double belts per inch of width, since it is found that under these mild conditions, leather belts run for many

years with a minimum of lubrication on the bearings, and with far less chance of heating the journals or stressing the shafts.

Power of Single Leather Belts.—To roughly calculate the power of single leather belts, the following formula may be used: Let H. P. = actual horse-power; W = width of belt; F = driving force; T = working tension from 70 to 150 lbs. per inch of width; V = velocity of belt in feet per minute. Then

$$F = \frac{33{,}000 \times \text{H.P.}}{V}, \quad \text{or} \quad F = \frac{W \times T}{2}. \quad \text{H.P.} = \frac{V \times F}{33{,}000}.$$

$$W = \frac{33{,}000 \times \text{H.P.}}{\frac{T}{2} \times V}.$$

These rules give good practical results where there is no great inequality in the diameter of the pulleys.

Table 26 will be found very useful in estimating the horse power that different sized belts will transmit. For instance a single oak tanned leather belt running at a speed of 700 feet per minute will transmit 1.06 horse power per inch in width, while a heavy double belt of the same material and at the same speed will transmit 1.91 H.P.

Which Side of the Leather Should Face the Pulley.—The grain or hair side of the leather is naturally much smoother than the flesh side, and millwrights differ in opinion regarding which of these sides should run in contact with the pulley. Whichever side of the belt is in contact with one pulley, the same side should

TABLE 26

HORSE-POWER THAT DIFFERENT LEATHER BELTS WILL TRANSMIT PER INCH IN WIDTH AT VARIOUS SPEEDS.

Velocity of Belt per Minute.	KIND OF BELTS.								
	Best Oak-tanned Belts.			Best Link or Chain Belts.					
	Single belts.	Light double belts.	Heavy double belts.	⅜″	½″	⅝″	¾″	⅞″	1″
Feet.	HORSE-POWER THEY WILL TRANSMIT.								
100	·15	·21	·27	·13	·15	·17	·20	·24	·27
200	·30	·42	·55	·25	·29	·35	·40	·47	·55
300	·45	·64	·82	·38	·44	·52	·60	·71	·82
400	·61	·85	1·09	·51	·58	·69	·80	·95	1·09
500	·76	1·06	1·36	·64	·73	·86	1·00	1·18	1·36
600	·91	1·27	1·64	·76	·87	1·04	1·20	1·42	1·64
700	1·06	1·49	1·91	·89	1·02	1·21	1·40	1·65	1·91
800	1·21	1·70	2·18	·92	1·16	1·38	1·60	1·89	2·18
900	1·36	1·91	2·45	1·05	1·31	1·55	1·80	2·13	2·45
1000	1·51	2·12	2·73	1·27	1·45	1·73	2·00	2·36	2·73
1100	1·67	2·33	3·00	1·40	1·60	1·90	2·20	2·60	3·00
1200	1·82	2·55	3·27	1·53	1·75	2·07	2·40	2·84	3·27
1300	1·97	2·76	3·55	1·65	1·89	2·25	2·60	3·07	3·55
1400	2·12	2·97	3·82	1·78	2·04	2·42	2·80	3·31	3·82
1500	2·27	3·18	4·09	1·91	2·18	2·59	3·00	3·55	4·09
1600	2·42	3·39	4·36	2·04	2·33	2·76	3·20	3·78	4·36
1700	2·58	3·61	4·64	2·16	2·47	2·94	3·40	4·02	4·64
1800	2·73	3·82	4·91	2·29	2·62	3·11	3·60	4·25	4·91
1900	2·88	4·03	5·18	2·42	2·76	3·28	3·80	4·49	5·18
2000	3·03	4·24	5·45	2·55	2·91	3·45	4·00	4·73	5·45
2100	3·18	4·45	5·73	2·67	3·05	3·63	4·20	4·96	5·73
2200	3·33	4·67	6·00	2·80	3·20	3·80	4·40	5·20	6·00
2300	3·49	4·88	6·27	2·93	3·35	3·97	4·60	5·44	6·27
2400	3·64	5·09	6·55	3·05	3·49	4·15	4·80	5·67	6·55
2500	3·79	5·30	6·82	3·18	3·64	4·32	5·00	5·91	6·82
2600	3·94	5·52	7·09	3·24	3·78	4·49	5·20	6·15	7·09
2700	4·09	5·73	7·36	3·28	3·85	4·66	5·40	6·38	7·36
2800	4·24	5·94	7·64	3·31	3·86	4·73	5·60	6·62	7·64
2900	4·39	6·15	7·91	3·32	3·87	4·78	5·80	6·85	7·91
3000	4·50	6·36	8·18	3·31	3·86	4·75	5·97	7·09	8·18
3100	4·60	6·58	8·45	3·30	3·85	4·73	5·96	7·33	8·45
3200	4·69	6·79	8·70	3·28	3·82	4·71	5·94	7·37	8·73
3300	4·77	7·00	8·86	3·24	3·77	4·70	5·92	7·35	8·88
3400	4·84	7·21	8·96	3·19	3·71	4·64	5·87	7·32	8·86
3500	4·90	7·31	9·06	3·13	3·61	4·50	5·78	7·26	8·80
3600	4·95	7·40	9·16	3·05	3·50	4·37	5·67	7·16	8·73
3700	4·99	7·48	9·24	2·96	3·39	4·26	5·55	7·01	8·58
3800	5·03	7·54	9·29	2·84	3·28	4·15	5·41	6·87	8·41
3900	5·03	7·60	9·34	2·72	3·13	4·02	5·20	6·70	8·27
4000	5·08	7·64	9·37	2·58	2·95	3·84	5·01	6·48	8·04

be in contact with all the pulleys over which it passes, so as not to bend it alternately in different directions. When the flesh side is next to the rim of the pulley, it should receive one coating of currier's dubbing, and three coatings of boiled linseed oil every year, in the case of a belt having to endure continuous hard work. This has the effect of rendering it about as smooth as the hair side, and its efficiency as a whole is said to be thereby increased. When the hair side is in contact with the pulley, it is usual to give that side an occasional coating of castor oil in order to render it more flexible, and, consequently, more durable than it would be in its natural state. Rosin or cobbler's wax should *never* be employed, as they gather dirt, and form lumps on the pulley and belt.

Double and Treble Belting.—Sometimes the breadth of a single belt necessary to transmit a given power would be inconveniently great, and hence double or treble belts are used. These are made by cementing, and then riveting, or sewing two, or three thicknesses of belting together. Owing to the greater rigidity of such belts they do not work so satisfactorily as single ones, and are not well adapted for running at high speeds, or over pulleys having a diameter of less than 3 or 4 feet, or in cases where the pulleys are close together.

Woven Belting.—Although leather belting is still very largely employed, it has, more especially during recent years, been very largely displaced by woven beltings, which are strong and very pliable—two of the most important features in any flexible transmitter of power—while they have the further advantage of not being affected by changes of temperature, water, steam, fumes of chemicals, etc.

Selecting Rubber Belts.—The two determining qualities in rubber belting are the weight of the cotton duck from which the belt is made and the weight and quality of the rubber coating. Rubber is becoming so scarce and so high-priced that hundreds of substitutes are used, and real rubber may soon become exceedingly scarce. Some rubber substitutes have the bad habit of peeling from the cotton as soon as the belt gets to working.

But in this matter there is another side in which the belt maker should be protected. Peeling and splitting of rubber belts sometimes take place when real rubber is used and the belt is first-class in every way.

FIG. 102a.

In cases like these, the belt maker should be protected against loading the belt to more than its safe working tension. It has already been stated that a belt which is not loaded to more than 40 pounds working pressure to the inch of width is not likely to slip or break. But the working strain is not by any means all the stress that comes upon a belt. A careless workman may put a belt on its pulleys in such a manner that it will not stand up under 40, or even under 20 pounds working pull. In fact, a belt may be placed upon pulleys so tightly laced or otherwise joined that it will not stand its own weight.

The determination of the lacing stress in belts will be discussed under the head of lacing, and is mentioned

here only to show that more belts fail on account of too tight lacing than fail on account of poor material being used in their making. The weight of rubber belts, and the thickness of the rubber coating, together with the weight of the cotton duck, are determining factors in selecting belts of this kind.

Finding Required Length of Belts.—When the pulleys are in place, the next step will be to ascertain the

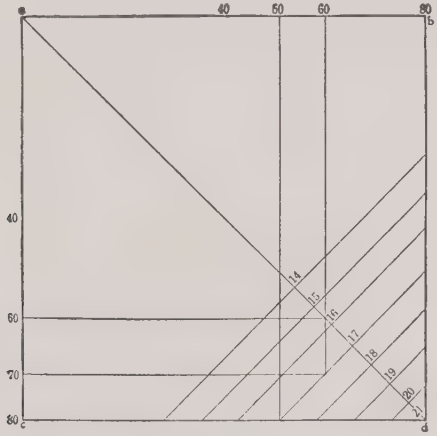


FIG. 102b.
LAYING OUT LENGTH OF BELT.

lengths of the various belts. A steel tape is without doubt the best device for measuring around pulleys. An ordinary tape line will answer fairly well. After the measurement has been taken, an allowance must be made for stretch of the belt when placed upon the pulleys. It is the usual custom to allow $\frac{1}{8}$ inch to the foot for stretch, therefore the belt should be cut 1 inch short for every 8 feet of its length. For instance, if the measurement on the pulleys showed 80 feet in

length, the belt should be cut $80 \div 8 = 10$ inches short, which would make its length about 79 feet. Another method of ascertaining the lengths of the belts to be used in the mill or factory under construction, is by means of a belt chart, see Figs. 102b and 102c. The distance between centers of the different shafts being known, a chart or diagram may be constructed very easily from which may be taken instantly the length of belt necessary to wrap half way around each of a pair of pulleys. While the chart does not take into account the slight difference due to the difference in arcs of contact of belts over pulleys of unequal diameter, the error thus caused is very slight, and is only a very few inches in extreme cases of a very large and a very small pulley.

For instance, should it be necessary to belt a pair of 80-inch pulleys the length of belt taken up by wrapping half way around each of the pulleys will equal the circumference of one of the pulleys, and this is $80 \div 12 \times 3.1416 = 20.93$ feet; so close that we will call it 21 feet. To originate the chart or diagram, draw the line a d, Fig. 102b, making it 21 parts long by any convenient scale. This line should be drawn at an angle of 45 degrees with lines a b and a c, which should then be divided into 80 equal spaces, each space representing 1 inch of pulley diameter.

The diagonal line a d should also be divided, but into 21 equal parts, each part representing one foot of belt. Through these points the lines 14 ,15, 16, etc., should be drawn and marked with the figures mentioned, commencing at a, which represents 0, the next division 1, the next 2, etc., the last division 21, being located at the intersection of lines d, b, and d, c.

Reference to Fig. 102c will show that by following

this method, a chart may be constructed for all pulleys up to 80 and 120 inches in diameter. The method of using the chart is as follows. It is desired to ascer-

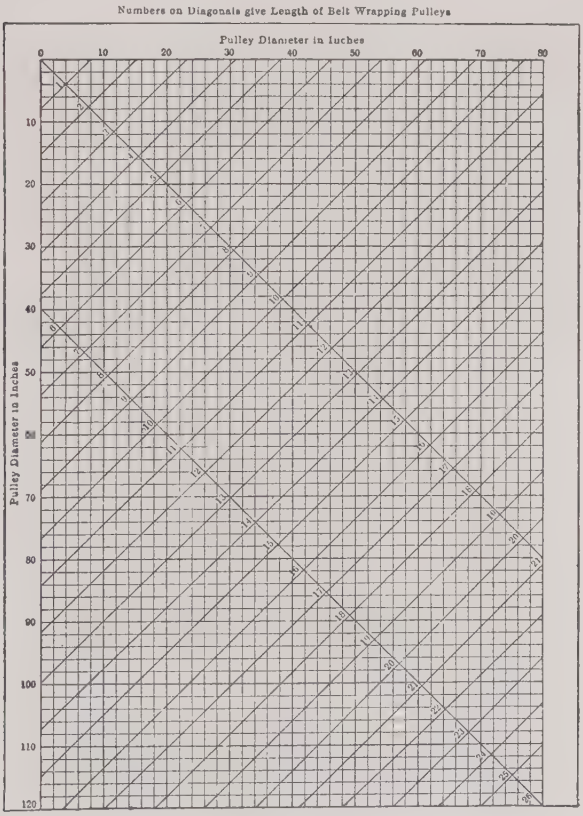


FIG. 102c.
CHART FOR FINDING LENGTH OF BELT.

tain the length of a belt required to connect two pulleys, having diameters of 36 inches and 48 inches respectively, and carried by shafts with distance between centers of

19 feet. The chart shows that pulleys of 36 and 48 inches in diameter require about 10 feet, 10 inches, or say 11 feet of belt wrap, and adding this to 38 feet which is twice the distance between centers, gives $11 + 38 = 49$ feet as the required length of belt.

Belt Fastenings.—The best fastening for a belt is the cement splice. It is much better than lacing, belt hooks, riveting, or any other method of joining together the ends of a belt. Good glue makes a strong and durable cement for leather belts, but fish glue is less affected by moisture. Ordinary hot glue may be made impervious to water by adding a small quantity of bichromate of potash to the glue just before using.

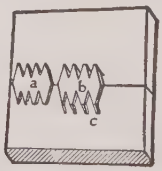


FIG. 102d.

Lacing Belts.—Joining the ends of a belt by lacing tends to weaken the belt by just the amount of material that is punched out for the holes that are required to receive the lacing. The best method to pursue in lacing is, to use as small lacing as possible, and pass it many times through small holes, rather than a wide lacing passed through large holes. As an instance of the weaking effect of punching holes of too large diameter in a belt for the purpose of lacing it, the following example may be cited. Taking a double belt 5 inches in width, although the width is immaterial as the loss of strength is calculated as a given per cent of total width. There are four holes in this piece of belt, each hole $\frac{3}{8}$ inch in diameter. The aggregate width

thus cut out of the belt is $4 \times \frac{3}{8}$ inches $= 12/8 = 1\frac{1}{2}$ inches. Then $1.5 \div 5 = 0.30$, or 30 per cent of the belt has been cut away—nearly one-third of the total strength.

Now if in place of a few large holes there are more smaller ones—one-fourth more, in fact. There are five holes, each 3/16 inch in diameter, making a total of 15/16 inches, or $0.9375 \div 5 = 18\frac{3}{4}$ per cent, leaving $81\frac{1}{4}$ per cent of the total belt strength against 70 per cent in the belt with large holes. A first-class double leather belt will tear in two under a strain of about 500 pounds to each lace hole, the strain being applied in the holes by means of lacings.

Thus it would require 2,500 pounds to tear the 5-inch belt with the small lace holes, while the belt with the large holes would only stand 2,000 poundts.

Another method of computing the working strength retained in the belt after punching the holes is, assuming the tensile strength of the leather to be 3,000 lbs. per sq. in., the 5-inch belt being $\frac{3}{8}$ in. thick will have a cross sectional area of $5 \times 0.375 = 1.875$ sq. in. After deducting 30 per cent of this for the holes, there still remains a belt section of 1.125 sq in., having a breaking strength of $1.125 \times 3{,}000 = 3{,}375$ lbs. But as the large lacing and lace holes of the belt broke it under a strain of 2,000 lbs., it is plainly evident the strain is not evenly divided. With the other method of lacing, viz: small holes, the belt retains $81\frac{1}{4}$ per cent of its original cross section $= 1.525$ sq. in. after punching the five small holes. The breaking strain in this case $= 3{,}000 \times 1.525 = 4{,}575$ lbs., using a factor of safety of 5 the working strain $= 4{,}575 \div 5 = 915$ lbs. per sq. in., or $1.525 \times 915 = 1{,}395.375$ lbs. for the belt. There being 5 lace holes, the pull on each lace $= 1{,}395.375 \div 5 =$

279 lbs., and using a factor of safety of 2 for the lacing, the safe working strength of which $= 500 \div 2 =$ 250 lbs., it will be seen that the stress is more evenly divided by the use of the small holes and a larger number of them.

Belt Hooks.—Fig. 102d shows a belt fastening which loses but very little belt section, the points of the fastenings being driven down between the fibers and cut but very few of them. One of the fastenings driven into place is shown at a, while another clip is in place at b all ready for the hammer, which is the only tool required for applying these fastenings. As may be seen at c, the points which go through the belt are very slim and quite small. They are made of rolled steel and will stand a heavy strain without breaking or bending.

These fastenings are made in various lengths and sizes to fit any and all belt thickness and width. After the hooks have been driven, the ends of the belt should, as shown in the engraving, lie evenly and close together. If the fastenings are driven properly, the belt-ends will be drawn closely together, but if the steel points are allowed to wander away from the end of the belt a little, the job will not be a good one. The points should project through the belt between 1/16 and 1/12 inch after having been driven through the belt into a piece of soft wood placed underneath. Then the belt should be turned over, placed upon the rim of an iron pulley, or a bar of iron or steel, and the points clinched and driven back into the belt. They should be driven in so far that they cannot touch the pulley at all, and upon the manner of the clinching and driving depends largely the value of this excellent fastening—a very desirable one when properly selected and applied.

Fig. 103 shows a variety of belt joints. At (1) is

shown a joint which is generally made in a belt factory where the necessary scarfing planes, and compressing gear are available. Cemented joints made in this way are nearly as strong as riveted joints. A good connector for new and strong leather belts is shown at

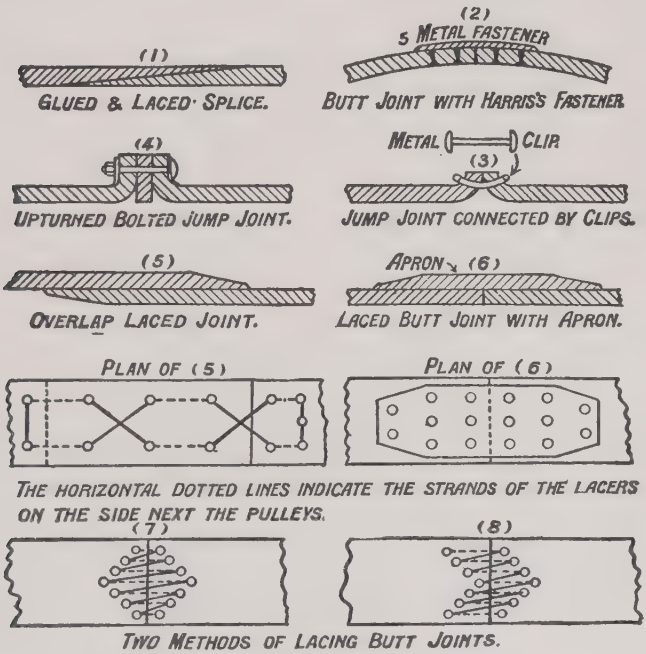


FIG. 103.
VARIOUS METHODS OF JOINING LEATHER BELTS.

(2). The tough yellow metal fasteners shown at (3) serve well for belts running at high speeds over small pulleys.

The slits for these clips are made by a special form of cutting pliers and should be cut fully ¼ inch from the

end of the belt, being so spaced that the crossheads of the fasteners come close to each other across the strap. In order to prevent the body of the metal clip from being bent up and down when passing over the pulleys, the fastener should be as short as possible.

At (4) is shown what is termed a "jump joint" upturned and bolted. The strength of this joint is

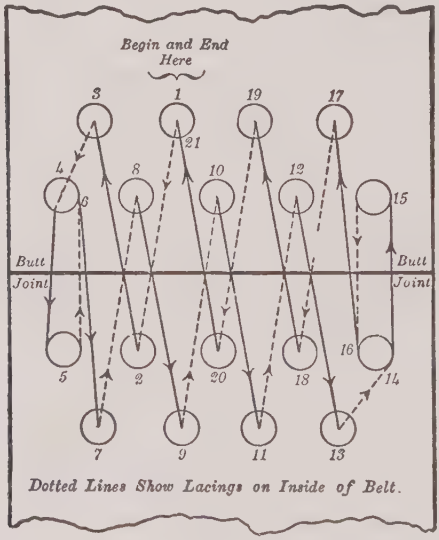


FIG. 104.
PLAN OF A LACED BUTT JOINT FOR 6-INCH BELT.

increased by inserting a leather strip or washer between the turned up ends as shown. An overlap lace joint in both section and plan is shown at (5). In making this joint, care should be taken to taper down and curve the ends to suit the smallest pulley over which the belt has to pass, otherwise the joint will be stiff, and consequently every time it travels on and

off the pulley a sort of hinge action takes place, accompanied by a shock which not only shortens the life of the belt, but also communicates a jar to the shaft and machine being driven.

For heavy broad double belts where a scarfed cemented joint cannot be used, the form of joint shown in plan and section at (6) with leather apron makes a very good connection; (7) and (8) show the best methods of lacing simple butt joints. The holes are punched so as to form a diamond or pointed figure,

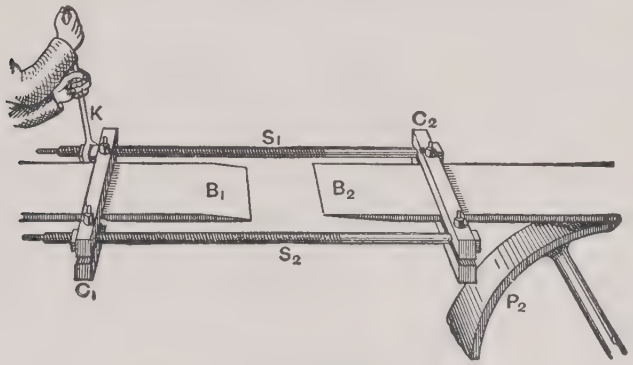


FIG. 105.
BELT STRETCHER.

whereby there are never more than two holes in line across the belt. Consequently it is possible to retain almost the entire strength of the belt without impairing its flexibility. Another plan of lacing a butt joint, which is very suitable for dynamo driving, and insures steady, smooth running, is illustrated in Fig. 104.

This joint is made by a lace in one length by beginning at hole 1, and continuing through the several holes in the order 2, 3, 4, etc., as shown in Fig. 104, ending with hole 1, which is also marked 21.

In order to facilitate the making of any one or other of the above-mentioned joints in belts more than 6 inches wide, it is usual to employ a "drawing-up frame" of the form shown by Fig. 105. The two ends, B_1, B_2, of the belt, after being passed over the driving and driven pulleys, are brought towards each other and gripped by the clamps, C_1, C_2. The wrench, K, is then applied alternately to the nuts of the long screws, S_1, S_2, until the tapers of the splice are brought fairly over each other, or until the ends come together fairly and squarely in the case of a butt joint. The lacing can then be done without any other effort than that of

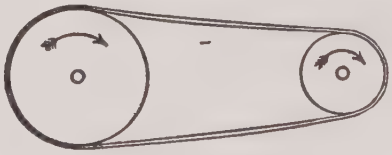


FIG. 106.
COMPOUND BELT DRIVING.

merely passing the thong through the holes and fastening it.

Compound Belting.—In order to avoid the internal straining action of thick double or treble belting, it has been found advisable to simply put two or three thin single belts on the top of each other, without any cementing or riveting together, as shown by Fig. 106. This gives perfect freedom of action to each belt to accommodate itself to the curves over which it passes without stressing its neighbours. The friction between the surfaces of these several belts is not found to have any observable objections, and it is said that 70 per cent more power can be transmitted by compounding two single belts in this manner.

By the addition of a third belt, still more power may be transmitted. A simple modification of this compound belt drive is illustrated by Fig. 107. The power is transmitted from one main driving pulley to two or more driven pulleys by separate belts, all moving in the same plane, and in a direct line with each other. This arrangement saves considerable room in a workshop or factory. It has been proved to be very convenient for driving several dynamos direct from one flywheel, instead of from independent drums.

Link Chain Belting.—This form of belting (see Fig. 108) permits of the escape of the air through the spaces between the links, and thus enables the leather to bear uniformly over the face of the pulley.

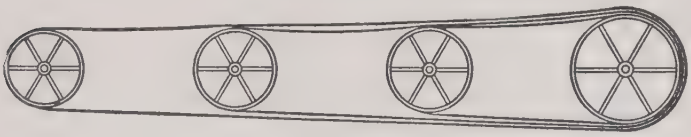


FIG. 107.
COMPOUND BELT DRIVE FOR THREE MACHINES.

It is composed of a series of short leather links bound together by steel pins and washers. These belts possess considerable flexibility, and can be made with a flexible central row of links to automatically suit the arch of any pulley. If desired, they may be specially curved to suit any camber. They further possess the advantage of being easily shortened and rejointed by simply bringing the ends together with a drawing-up frame, interlocking the links until a row of rivet holes are brought fair in a line, and then inserting the rivet. When one side of this belting is made thicker than the other it is well adapted for working on tapered cones and for quarter twist driving.

Creep of Belts.—Owing to the elasticity of belts, there is a certain amount of movement of every belt in a direction opposite to the motion of the pulley. This movement, called the "creep" of the belt, is usually taken at 2 per cent of the velocity of the driving pulley. Referring to 109, when the belt approaches the

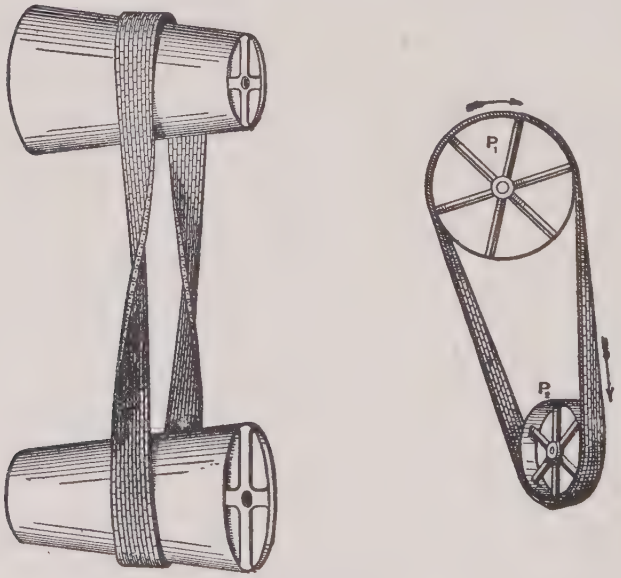


FIG. 108.
CHAIN BELTING—CHAIN BELTING WORKING QUARTER TWIST.

driven pulley B, it is under the least tension, and does no actual work, except that of holding itself against the stress of its own weight while suspended between points a and h. The arrows show the direction in which the belt is running. A is the driving pulley. If marks are made at a on the belt, and at b on the rim of driven pulley B exactly opposite mark a, and the pul-

ley be then revolved by the belt, it will be found that when mark a left the pulley it would no longer coincide with mark b, but would be some distance ahead of it, as shown at c and d. The cause of this is that when the pull exercised by pulley A became felt by the belt as it passed around pulley B, the belt was stretched a certain amount, being thus pulled slightly ahead of mark b on pulley B. This amount is usually about 1 per cent of the distance traveled by the belt. At driving pulley A, another lag occurs. It will be noticed that marks e and f on belt and pulley respectively coincide at the bottom, but as the pulley revolves drag-

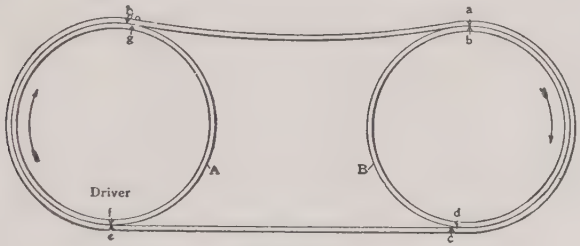


FIG. 109.
THE CREEP OF A BELT.

ging the belt with it to point g, the tension upon the belt is released, and the elasticity of the belt causes it to shorten to its normal condition again, creeping from g to h. In this case, the belt has fallen behind the pulley to the amount of 1 per cent of the distance traveled by the belt. In both cases the loss is about the same, resulting in a decrease of 2 per cent in speed of driven pulley, as for instance, if the driven shaft B were required to run at exactly 100 revolutions per minute it would be necessary to give the driving shaft A a speed of 102 revolutions per minute in order to make up for the creep of the belt in the manner described.

Thus, for a shaft belted 7 removes from the prime mover, the speed of that mover would have to be increased nearly 15 per cent (14.86) to make up for the creep of belts.

The above allowance is a very important one when exact speeds are required, and it explains why, in some cases, the actual speed does not come up to the requirements, even though the pulleys all figure correctly to give the required speed.

Rope Transmission.—The transmission of power is frequently accomplished by means of ropes instead of belts. In addition to other advantages of this method, ropes seldom give way without due warning by slackening or fraying in one or more of the strands, thus reducing the risk of accident, and stoppage of the mill or factory to a minimum. The working stress of the ropes being but a small fraction of their breaking strength, any signs of weakness in an individual rope would allow it to be removed, and the engine run with the remaining ropes until a convenient opportunity is offered for the replacement of the weak member. Further their comparative slackness between the pulleys facilitates their annulling any small irregularity in the motive power. These ropes are made of manilla-hemp, cotton, leather, and raw ox hide. Hemp ropes are preferred to cotton ropes for main drives with large pulleys, since they are cheaper, stronger, and last nearly as long as if spun with a greased core.

Cotton ropes, being softer and more pliable than manilla ropes, can be used with smaller pulleys without undue injury to the fibres. This is also considerably aided by the natural wax in the structure of the long staple variety which acts as a lubricant, and permits of greater freedom of motion between the several

fibres. The life of a good cotton rope is usually about thirteen years if properly adjusted and well treated.

Sizes of Ropes and Pulleys.—The size and number of ropes, as well as the least diameter of pulley, for any given power, are points of importance and should be considered for each case.

The ropes commonly used for the transmission of power in factories or mills vary from 3 to 5 inches in circumference. No matter what the diameter of the pulley may be, ropes of 1¾ inches diameter should not be exceeded for main drives, and 1¼ inches diameter for secondary drives. The diameter of the smallest pulley should not be less than thirty times the diameter of the rope, as the larger the pulley the less will be the internal friction, and consequent injury to the rope from bending and unbending (see table 27).

Strength of Cotton and Hemp Ropes.—The ultimate strength of white untarred cotton ropes may be taken at 9,000 lbs. per square inch of the net sectional area, which is about 90 per cent of the area of the circumscribing circle. The normal working stress should not exceed 1/30 of the ultimate breaking stress, or say 300 lbs. per square inch, although ropes are frequently worked at even a less tension.

Table 27 gives useful data regarding these points. The ultimate strength of *new manilla* ropes is about 11,000 lbs. per square inch of the net sectional area, which in a three stranded hawser is only about 80 per cent of the arc of its circumscribing circle. The necessary lubrication, however, reduces the strength by 20 to 30 per cent., but the lubrication of the fibres is of much greater importance than the actual breaking stress. The greater freedom of movement amongst the fibres permits a heavier working stress to be carried, and

ensures a much longer life. A properly lubricated manilla rope will outlast from two to four similar dry laid ropes working under the same conditions. There are many ways of lessening the internal friction, one of the best being that of coating the several yarns with a mixture of black lead and tallow prior to twisting the same into strands. Under favorable conditions, when thus treated, it is practically waterproof, and will last for about eight years.

TABLE 27.—COTTON DRIVING ROPES AND PULLEYS.

Diameters. Ins.	Areas. Ins.	Weight per Foot. Lbs.	Ordinary Working Tension including Centrifugal Force. Lbs.	Velocity of ropes, 4,700 ft. per minute (their most effective speed).			Rope Pulleys.	
				Stress due to Centrifugal Force Lbs.	Effective Tension Lbs.	Power each Rope will Transmit H.P.	Centre to centre of ropes Ins.	Diam. of smallest pulley Ins.
$\frac{1}{2}$	.1963	.081	47	16	31	4.43	$\frac{7}{8}$	15
$\frac{5}{8}$	.3067	.125	72	24	48	6.84	1	18
$\frac{3}{4}$	.4417	.184	106	35	71	10.07	$1\frac{1}{8}$	22
$\frac{7}{8}$	.6013	.25	144	48	96	13.67	$1\frac{5}{16}$	26
1	.7854	.33	190	63	127	18.05	$1\frac{1}{2}$	30
$1\frac{1}{4}$	1.2272	.51	294	98	196	27.90	$1\frac{13}{16}$	37
$1\frac{1}{2}$	1.7671	.74	426	142	284	40.48	$2\frac{1}{8}$	45
$1\frac{3}{4}$	2.4053	1.00	576	192	384	54.70	$2\frac{1}{2}$	52
2	3.1416	1.30	750	250	500	71.10	$2\frac{3}{4}$	60

Splicing Ropes.—The splicing of ropes is a matter of great importance, and should never be left to unskilled hands. In order to secure a sufficiently strong joint its length requires to be from forty to fifty times the diameter of the rope, or say from 6 to 7 feet for main, and from 4 to 5 feet for secondary drives. The different strands should be neatly interlocked, so that the thickness of the rope is not increased. The strength of a good splice is only about 70 per cent of that of the rope.

Rope Pulleys.—The pulleys used for rope gearing are made with V-shaped grooves around their rims.

The sides of the grooves usually make an angle of 45° with each other. The ropes must never be allowed to rest on the bottom of the grooves, but only on the sides, as shown. They are thus wedged into the grooves, and the resistance to slipping is thereby greatly increased. This has the effect, however, of producing greater wear and tear of the ropes. In the case of guide pulleys the rope should always rest on the bottom of the groove. In the case of belts running on flat-rimmed pulleys, sufficient adhesion to prevent slipping of the belt on the pulleys is obtained by stretching the belt over the pulleys with an initial tension. But with ropes it is found that smoother working, and less wear and tear take place when the rope is put on the pulley with as little initial stress as possible. Hence the reason for increasing the frictional resistance between the rope and its pulleys by allowing the rope to wedge itself into the grooves in the pulley rim.

The necessary pressure between the rope and the sides of the grooves to ensure sufficient frictional adhesion at the commencement of motion, is simply that due to the weight of the hanging parts of the ropes between the pulleys. The pulleys should, therefore, be large, and be placed at a sufficient horizontal distance apart, so as to have the arcs of contact between the rope and the pulleys as great as possible.

Fig. 110 shows the construction of a large rope flywheel, in which the wrought-iron bolt, B, connects the grooved rim, G R, with the wheel boss, W B, and thus receives the tensile stress due to centrifugal action on G R.

In horizontal drives, the tight side of the rope should always be in contact with the lower parts of the pulleys, and the slack side above, so as to obtain a maximum

arc of contact between the rope and the pulleys. The groove action of rope pulleys also acts in opposition to centrifugal force, which latter is so potent a factor in belt-driving.

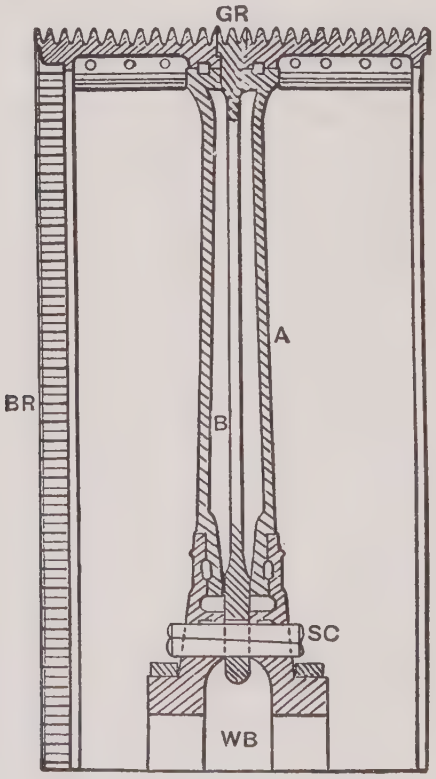


FIG. 110.

SECTION OF LARGE ROPE PULLEY.

Index to parts.—WB, wheel boss; SC, steel cotters; B, wrought iron bolt; A, arm; GR, grooved rim; BR, barring rack.

Ropes can therefore be run at a much higher velocity than belts. Although the usual velocity of rope trans-

mission does not exceed 5,000 feet per minute, yet there are cases where satisfactory results are being obtained from rope transmission at a speed of over 7,000 feet per minute, and the claim is made that the ropes last as long as when running at 5,000 feet per minute. The following formula provides an ample, though not extravagant margin of safety against collapse to individual ropes in fair-sized installations, and also conforms to the regular findings of engineering practice:

$$\text{H. P.} = \frac{\text{V}\ (24.6\text{d} - 12)}{4860 - 1000\ \text{d}},$$

in which H. P. = horsepower transmitted by good 3-strand cotton driving ropes running on pulleys having diameters not less than 30 times the diameter of rope, V = velocity of rope in feet per minute, and d = diameter of rope in inches. (Arc of contact of rope and pulleys assumed as 170°.)

With regard to the relative cost of ropes and of belts, practical experience has shown a very great advantage in favor of the former, and the advantage is more marked in the case of thin ropes, say 13/16-in. in diameter, such as are used for driving light machinery, like ring spinning-frames.

Besides the cases mentioned, there are many other directions in which rope driving may be adopted with advantage, particularly where the electrical transmission of power is concerned, for coal-washing machinery, for colliery fans, brick-making machinery, and the like.

Casing of Pulleys.—An important consideration often omitted when arranging for rope driving is, casing the sides of pulleys with sheet-metal or boards, without

which the displacement of air by the arms of large high-speed pulleys adds materially to the profitless load.

Minimum Diameter of Smallest Pulley.—The item which appears most naturally to offer itself first for consideration, is the bending, and consequent gripping capability of the rope itself. Soon after cotton came into use for main driving ropes it was discovered that a three-strand rope of this material would cling firmly to a pulley thirty times its own diameter, but would gradually relax its hold as the relative diameter decreased from this standard. Subsequent practice places the validity of this conclusion beyond the possibility of doubt.

Proportionate Durability of Ropes on Large and Small Circumferences.—Too much importance can scarcely be attached to the relative proportions of pulleys and ropes, for it is absolutely certain that both driving force and durability suffer most materially from the non-observance of the thirty-diameter regulation, and that nothing, not even space restriction, has gone so much against the successful application of ropes to the driving of dynamos, or transmission from high-speed motors, as a persistent effort to get below this natural limitation. Therefore, instead of adding to the burden of the drive by employing an abnormal quantity of thick ropes to make up for grip depreciation when large pulleys prove impracticable, it is far better to divide the total bulk nominally required into the necessary number of smaller ropes.

Arc of Contact.—Before dismissing the consideration of pulley diameters it will be well to notice another phase of the same question—viz., that which deals with the arc of contact. It needs but little reasoning to prove that rope contact reaches its maximum when both

pulleys are equal in size; but when unequal the utmost quantity realizable is that carried by the smallest pulley, whether it be driver or driven.

Trailing Span of Ropes.—The arc of contact is also affected by the position occupied by the trailing span of the ropes, which, when above the pulleys, undoubtedly gives the maximum bearing surface. This position is not always attainable, nor is it under all circumstances desirable. For example, an erratic delivery of power or variable load may set up oscillations in the responsive ropes of such violence as to fling them entirely off the pulleys. Now if the direction of rotation is so arranged as to bring the trailing span below the pulleys, any tendency on the part of the ropes to wander from their appointed track is held in check by the pull on the right side, while gravitation tends to control the vibrations on the slack side.

Loss due to reverse driving may for safety be calculated as the difference between the average of 170° and the arc of actual contact, which is usually measured from the tangent point of a curve whose depth (from a straight line drawn along the bottom of the pulleys) is one-tenth that of the distance from center to center of the shafts and the taut portion of the rope stretched across the top of the pulleys. With oblique driving the same tendencies do not prevail.

Angular Driving.—There is no class of power transmission which lends itself better to the treatment of what may be considered awkward conditions than rope driving. With the aid of guide pulleys any angle may be negotiated or corner turned.

Shafts Slightly Out of Alignment.—When shafts are only slightly out of alignment ropes will direct themselves without the aid of guide pulleys, but to what

extent this angularity may be carried depends largely upon center distances and pulley diameters. For example, under what may be considered good conditions —say with centers not less than three times the average diameter of the pulleys—a deflection of 7° might be ventured upon. Should the angle of deflection from the parallel prove too great to be treated in the above mentioned manner, the drive may be split up into two or more spans by the introduction of intermediate shafts and the angle divided between them. An angle of 15° was thus successfully negotiated. It should be stated that the grooves in all the pulleys were turned to an angle of 50°, and rather more rope applied than would be considered necessary under ordinary conditions.

Grooves.—The fact that when loads are constant the trailing span curves by its own weight down upon or away from the pulleys, according to the direction of rotation, and is ejected, so to speak, without any disturbance or apparent effort, also that a cotton rope may be pressed as tightly as possible round the entire circumference, and will yet fall away immediately the ends are liberated, pretty conclusively demonstrates the futility of seeking easement in curvatures to match in some measure the roundness of the rope itself.

Angular Grooves.—Curves being disallowed, there remains only the angular groove. Undoubtedly, every rope has its most suitable groove angle, but as the providing of a range of templates to agree with every advance in size would, perhaps, prove more inconvenient than advantageous, two leading angles have been fixed upon for all driving ropes, viz., 40° for those above 1¼ ins. in diameter, and 30° below that size; excepting, however, the small bands used for driving cotton machinery—say, ½ in. in diameter and even less, when

angles as sharp as 15° are brought into requisition. In setting out a groove it should always be borne in mind that a rope is to all intents and purposes an elastic wedge, and reaches its highest driving capabilities when it is compressed to the shape of the groove itself. This form ought, therefore, always to be anticipated, for if the sides of the groove are merely shown as tangents to the circle representing a given diameter of rope, there is always the danger of making the groove too shallow, so that when compression takes place the rope is liable to find its way to the bottom, or, what is almost as detrimental, work upon the semi-circle with which grooves generally terminate, when grip is considerably weakened and slipping supervenes, the true indication of which is heated pulley-rims.

Flangeless Grooves.—Finding that superposed flanges do not contribute in any way to better transmission, but rather prevent the rope falling direct into its working position if the driving is at all unsteady, besides crippling any extension in the direction of a larger diameter, many manufacturers are now adopting flangeless grooves, which are set out in precisely the same manner, only that the angles are extended and simply rounded off at the terminals.

Driving Ropes.—Many and various are the materials which have been pressed into the service of the rope-maker—an almost infinite variety of fibers, through the whole range of hemps and flaxes, from the coarse-grained manila to the silk-like ramie, and finally cotton.

Manila.—Owing, it may be conjectured, to the fact that rope driving was cradled in what may be styled the home of the manila trade, that material gained for a time precedence above all others, and, if merely judged on the lines of resistance to tensile strains, would still

hold the field. The fibers are, however, of so harsh and wiry a nature that they cannot be compressed into a solid mass by the successive twisting operations to which they are subjected, but, instead of forming a perfect impact, the strands merely rest upon each other, thus constructing what may be termed a spiral tube. The strands therefore scrub one upon another with every bend of the rope until the center is reduced to a fine powder.

The permanent set is thus quickly reached, while gripping power is so limited that any reduction in the pulleys from 45 diameters must be reciprocated with a corresponding increase in the amount of rope.

Construction of Ropes.—Much depends upon the construction of a driving rope. Fibrous ropes are held together by a series of twisting operations from the fiber to the yarns, and then the strands, which are locked together by a reversal of the same agency. Ropes of three, four, and even seven strands are used.

A study of a section of three-strand rope discloses, among other advantages, the effect of triangulated strains upon pliable material under the equal pressure exerted by the twisting operation already mentioned. It is in fact a trinity of elastic, spiral wedges, so impacted that no space remains unoccupied, and so supple that they will bend to and fro without disturbing their mutual relationship. This venerable principle as applied to rope making, holds as good to-day as ever, and so far admits of no improvement, except probably in a few minor details.

Lubrication of Driving Ropes.—The chief function of lubricating material as applied to cotton driving ropes, is to provide a smooth, filmy coating to assist the lay, and prevent friction between the twisted surfaces in

contact where the strands impact with each other. The lubricant should, however, be applied sparingly, and only at long intervals, unless the ropes are working in an unusually dry atmosphere. Greasy compounds should be avoided, as they will tend to induce slipping. When ropes are inclined to slip, due to an over-dose of grease, a liberal application of whiting will absorb the grease, and then fall away in flakes.

A good lubricant for rope drives is composed of saponified tallow, wax and plumbago.

WHEEL GEARING.

General.—Toothed wheels are employed where motion has to be transmitted with accuracy. The teeth should be formed so that the friction between them shall be the least possible. The diameter of the wheels is measured at the pitch circle. The distance from the center to center of the teeth measured along the pitch circle is called the circular pitch, or commonly the "pitch." If D be the diameter of the pitch circle of a wheel, N the number of teeth, and P the pitch of the teeth, then $D \times 3.1416 = N \times P$. The "diametral" pitch of a wheel (used chiefly in smaller sizes) is the length bearing the same proportion to the circular pitch that the diameter of a circle bears to the circumference. Therefore if P is the circular pitch, p diametral pitch, D diameter of pitch circle, and N the number of teeth, we have:

$$\text{Circular pitch} \ldots\ldots\ldots = P = \frac{3.1416\ D}{N} = 3.1416\ p$$

$$\text{Diametral pitch} \ldots\ldots\ldots = p = \frac{P}{3.1416} = \frac{D}{N} = 0.318\ P$$

$= 0.318\ P$

$$\text{Diameter of pitch circle} \ldots = D = N\,p = \frac{P \times N}{3.1416} = 0.318PN$$

$$\text{Number of teeth} \ldots = N = \frac{3.1416\,D}{P} = \frac{D}{p}$$

The following table gives the diametral pitches corresponding to circular pitches from ½ in. to 12 in.:

TABLE 28.

Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.
1/2	0.1592	2 1/4	0.7162	5	1.5916
5/8	0.1989	2 1/2	0.7958	5 1/2	1.7507
3/4	0.2387	2 3/4	0.8754	6	1.9099
7/8	0.2786	3	0.9549	6 1/2	2.0691
1	0.3183	3 1/4	1.0345	7	2.2283
1 1/8	0.3581	3 1/2	1.1141	7 1/2	2.3875
1 1/4	0.3979	3 3/4	1.1936	8	2.5465
1 3/8	0.4377	4	1.2732	9	2.8647
1 1/2	0.4775	4 1/2	1.3528	10	3.1829
1 3/4	0.5570	4 1/2	1.4324	11	3.5014
2	0.6366	4 3/4	1.5120	12	3.8200

The so-called diametral pitch adopted by Messrs. Brown and Sharpe, is the reciprocal of the foregoing, or $\frac{1}{p}$.

For this method of computing the diametral pitch, the following tables will be of service. The first gives the equivalent diametral pitches for circular pitches of from 2 to $\frac{1}{16}$ in.; and the second the equivalent circular pitches for diametral pitches of from ¾ to 60.

TABLE 29.

Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.
2	1.5708	1	3.1416	1/3	9.4248
1 7/8	1.6755	15/16	3.3510	5/16	10.0531
1 3/4	1.7952	7/8	3.5904	1/4	12.5664
1 5/8	1.9333	13/16	3.8666	2/9	14.1372
1 1/2	2.0944	3/4	4.1888	1/5	15.7080
1 7/16	2.1855	11/16	4.5696	3/16	16.7552
1 3/8	2.2848	5/8	4.7124	1/6	18.8496
1 5/16	2.3936	9/16	5.0265	1/7	21.9911
1 1/4	2.5133	1/2	5.5851	1/8	25.1327
1 3/16	2.6456	7/16	6.2832	1/9	28.2743
1 1/8	2.7925	3/8	7.1808	1/10	31.4159
1 1/16	2.9568	5/16	8.3776	1/16	50.2655

TABLE 30.

Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.	Diametral Pitch.	Circular Pitch. in.
$\frac{3}{4}$	4.1888	8	0.3927	26	0.1208
1	3.1416	9	0.3491	28	0.1122
$1\frac{1}{4}$	2.5133	10	0.3142	30	0.1047
$1\frac{1}{2}$	2.0944	11	0.2856	32	0.0982
$1\frac{3}{4}$	1.7952	12	0.2618	34	0.0924
2	1.5708	13	0.2417	36	0.0873
$2\frac{1}{4}$	1.3963	14	0.2244	38	0.0827
$2\frac{1}{2}$	1.2566	15	0.2094	40	0.0785
$2\frac{3}{4}$	1.1424	16	0.1963	42	0.0748
3	1.0472	17	0.1848	44	0.0714
$3\frac{1}{2}$	0.8976	18	0.1745	46	0.0683
4	0.7854	19	0.1653	48	0.0654
5	0.6283	20	0.1571	50	0.0628
6	0.5236	22	0.1428	56	0.0561
7	0.4488	24	0.1309	60	0.0524

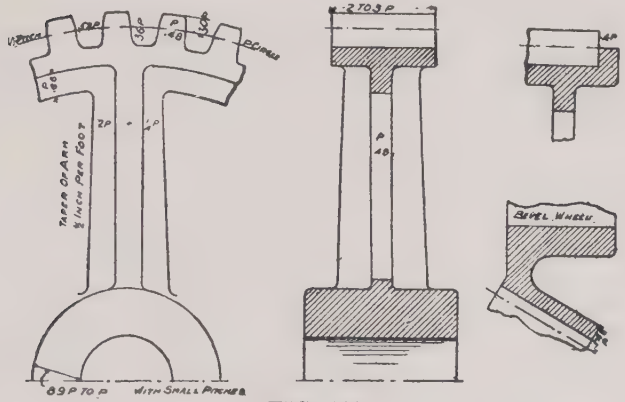


FIG. 111.

Proportions of Teeth.—The ordinary proportions of the teeth and arms of spur and bevel wheels in terms of the pitch are given in Fig. 111. In this case:

Thickness of teeth = 0.48 pitch.

Space between teeth = 0.52 pitch.

Length of teeth from pitch line to point = 0.3 pitch.

Length of tooth from pitch line to bottom of tooth = 0.4 pitch.

The following proportions are given by Unwin as preferable to the foregoing:

Thickness of tooth....$= 0.47\ P - 0.02$ to $0.48\ P - 0.03$
Width of space.....$= 0.53\ P + 0.02$ to $0.52\ P + 0.03$
Side clearance .....$= 0.06\ P + 0.04$ to $0.04\ P + 0.06$
Height above pitch line. $=$ $0.3\ P$ to $0.35\ P$
Depth below pitch line. $= 0.35\ P + 0.08$ to $0.4\ P + 0.08$
Total height of teeth..$= 0.65\ P + 0.08$ to $0.75\ P + 0.08$

The length of teeth is a matter in which much variation exists in practice, the present tendency being to adopt shorter teeth than in the older practice. Though this diminishes wearing surface, the teeth are less liable to fracture, which is an important point in high-speeded and heavily-stressed wheels. The old proportions for depth of teeth, including bottom clearance, gave 12/15 of the pitch. With high-speed wheels doing heavy work, this is too much. The proportion, 7/10, is better. During recent years several engineers have recommended as a general rule for heavy mill wheels, that the depth should not exceed one-half the pitch.

Helical Wheels.—Helical wheels are a legitimate development of the stepped gears, the object of which is to diminish the sliding action of the teeth. Thus in a stepped gear the total arc of contact, which measures the amount of sliding, is divided between several short teeth, and the sliding on each is reduced to

$$\frac{\text{Arc of contact}}{\text{number of teeth}}.$$

A helical tooth is simply one in which the steps merge into a spiral line, and as the points of contact change continually along the entire tooth plank, there is no sliding at all, but simply rolling. The double helical teeth are employed in order to balance the lateral forces which exercise undue pressure, tending to force the teeth away from contact in single helical wheels.

Spiral Gears.—Spiral gears bear a family resemblance to the helical gears. The difference lies in the direction of their axes. They are both alike screw-gears, and must be so designed. But while the action of helical wheels is entirely rolling, that of screw-gears is wholly sliding. The difference is due to the different positions of the wheel axis, which in the first case are parallel, and in the second at right, or other angles.

Worm-Gears.—Worm-Gears are particular applications of screw-gears. The worm is a wheel having a single thread, or tooth of very short axial pitch. The teeth of the wheel in which it gears are short sections of threads of extremely long axial pitch.

As in screw-gears, the action is wholly sliding. Double and treble threaded worms contain two and three screw threads. The axial pitch of the wheel teeth is much shorter, and the angle, or slope, greater than in those which gear with a single threaded worm. That the wheel teeth are short lengths of spirals of very long axial pitch is clearly evident in the case of straight wheel teeth of uniform section set diagonally on the rim, but is not so clear in the case of teeth which are cut to envelop the worm tooth. The latter is the proper form for the best results, especially when heavy gearing is in question, though light wheels of small pitch may be formed by the first method.

Wheel Arms.—The shapes of arms of wheels are settled more by convenience of manufacture than by any hard and fast rules. The strength is generally much in excess of that of the teeth, in order to absorb vibration. The arms are of T or ═, or 0 sections, the latter being chiefly used on change gears. T and + are employed mostly in pattern wheels, and ═ in those

made by machine, while the simple disc is reserved for many wheels in small diameter.

Number of Arms of Spur Wheels.—Under 3 ft. diameter should be 4; from 3 ft. to 8 ft., 6 arms; from over 8 ft. to 16 ft., 8 arms; from over 16 ft. to 24 ft., 10 arms.

Wheel Bosses.—Not the least important part of a wheel is its boss. These often cause fracture in consequence of being too large. The boss should never be heavy, especially when adjacent arms are light, and extra metal around keys should be provided.

Strength of Spur-wheel Teeth.—Innumerable rules have been formulated for the strength of the teeth of spur wheels, many of which give widely different results. Much of the existing confusion is due not only to the various estimates of the strength of the material, and the allowable factor of safety, but also to the manner in which the load is taken to fall on the teeth. It is frequently assumed that the tooth is liable to fracture across the corner, in which case only a portion of the breadth of the tooth equal to the length $\times \frac{1}{\sqrt{2}}$ ought to be considered as available. No doubt this condition may be justly assumed with roughly-cast gearing, or when the axes of the wheels are not exactly parallel or the bearings rigidly fixed, as well as in the case of crane gearing, or in gearing subject to shocks or rough usage. For well-made machine-moulded, or cut wheels, such as are now very generally used for main-driving gear, however, the teeth may reasonably be assumed to have a fairly even bearing throughout the whole of their length.

Taking a tooth of ordinary proportions, and making

TABLE 31.—HORSE-POWER OF SPUR WHEELS.
(Per Revolution per Minute and per Inch of Breadth.)

Pitch in Inches.	Diameter of Wheel.																					
	ft in 0 6	ft in 0 9	ft in 1 0	ft in 1 3	ft in 1 6	ft in 1 9	ft in 2 0	ft in 2 3	ft in 2 6	ft in 2 9	ft in 3 0	ft in 3 6	ft in 4 0	ft in 4 6	ft in 5 0	ft in 5 6	ft in 6 0	ft in 7 0	ft in 8 0	ft in 9 0	ft in 10 0
1	.006	.009	.012	.015	.018	.021	.024	.027	.030	.033	.036	.042	.048	.054	.060	.066	.072	.084	.096	.108	.120
1¼	.007	.011	.015	.019	.022	.026	.030	.034	.037	.041	.045	.052	.060	.067	.075	.082	.090	.105	.120	.135	.150
1½	.009	.013	.018	.023	.027	.031	.036	.040	.045	.050	.054	.063	.072	.081	.090	.099	.108	.126	.144	.162	.180
1¾	.010	.015	.021	.026	.032	.036	.042	.047	.052	.058	.063	.073	.084	.094	.105	.115	.126	.147	.168	.189	.210
2	.012	.018	.024	.030	.036	.042	.048	.054	.060	.066	.072	.084	.096	.108	.120	.132	.144	.168	.192	.216	.240
2¼	....	....	.027	.034	.040	.047	.054	.061	.067	.073	.081	.094	.108	.121	.135	.148	.162	.189	.216	.243	.270
2½	....	....	.030	.038	.044	.052	.062	.068	.074	.082	.090	.104	.120	.134	.150	.164	.180	.210	.240	.270	.300
2¾	....	....	....	.041	.049	.053	.066	.074	.082	.091	.099	.115	.132	.148	.165	.181	.198	.231	.264	.297	.330
3	....	....	....	.045	.054	.063	.072	.081	.090	.099	.108	.126	.144	.162	.180	.198	.216	.252	.288	.324	.360
3¼	....	....	....	.049	.058	.068	.078	.088	.097	.107	.117	.136	.156	.175	.195	.214	.234	.273	.312	.350	.390
3½	....	....	....	....	.064	.072	.084	.094	.104	.116	.126	.146	.168	.188	.210	.230	.252	.294	.336	.378	.420
3¾	....	....	....	....	.067	.078	.090	.101	.112	.124	.135	.157	.180	.202	.225	.247	.270	.315	.360	.405	.450
4	....	....	....	....	....	.084	.096	.108	.120	.132	.144	.168	.192	.216	.240	.264	.288	.336	.384	.432	.480
4¼	....	....	....	....	....	.089	.102	.115	.127	.140	.153	.178	.204	.230	.255	.280	.306	.357	.408	.460	.510
4½	....	....	....	....	....	.094	.108	.122	.134	.148	.162	.188	.216	.242	.270	.296	.324	.378	.432	.486	.540
4¾	....	....	....	....	....	....	.114	.128	.142	.157	.171	.199	.228	.256	.285	.313	.342	.399	.455	.512	.570
5	....	....	....	....	....	....	.120	.135	.150	.165	.180	.210	.240	.270	.300	.330	.360	.420	.480	.540	.600
5½	....	....	....	....	....	....	....	.148	.164	.182	.198	.230	.264	.296	.330	.362	.396	.462	.528	.594	.660
6	....	....	....	....	....	....	....	.162	.180	.198	.216	.252	.288	.324	.360	.396	.432	.504	.576	.648	.720

a suitable allowance for wear, we have, if the stress on the material is taken at about 1,500 lbs. per square inch,

$$\text{H. P.} = \frac{\text{D p b N}}{1000},$$

in which H. P. = horsepower; D the diameter of the wheel in inches; p the pitch; b the breadth; and N the number of revolutions per minute. It is assumed that one-half the load falls on each tooth, and that the teeth bear uniformly throughout their whole breadth. Table 31, based on these assumptions, gives the horsepower per revolution and per inch of width of spur wheels.

Weight of Spur-wheel Castings.—

$$W = \frac{P^2 \times N \times B}{C},$$

where P = pitch in inches; N = number of teeth; B = breadth in inches; W = weight in cwt.; C = constant, which up to 2 in. pitch = 270; from 2 in. to 3 in. = 250; from 3 in. to 4 in. = 245. Roughly, the weight of an iron casting equals fourteen times the weight of pattern.

Wheel-gearing Constants.—Table 32 of constants (C) will be found of service in calculations involving the pitch P, number of teeth N, and radius of pitch circle R, of spur wheels.

$$R = C \times P. \qquad P = \frac{R}{C}. \qquad C = \frac{R}{P}.$$

Examples.—Find the radius of pitch circle of a spur wheel having 45 teeth 1¾-in. pitch.

At the intersection of lines 40 and 5 in Table 32, we find the constant 7.16. Then $7.16 \times 1.75 = 12.53$, or 25 in. diameter.

TABLE 32.—TABLE OF WHEEL-GEARING CONSTANTS.

N	0	1	2	3	4	5	6	7	8	9
0	0.00	0.159	0.318	0.477	0.637	0.796	0.955	1.114	1.273	1.43
10	1.59	1.75	1.91	2.07	2.23	2.39	2.55	2.71	2.86	3.02
20	3.18	3.34	3.50	3.66	3.82	3.98	4.14	4.30	4.46	4.62
30	4.77	4.93	5.09	5.25	5.41	5.57	5.73	5.89	6.05	6.21
40	6.37	6.53	6.68	6.84	7.00	7.16	7.32	7.48	7.64	7.80
50	7.96	8.12	8.28	8.44	8.59	8.75	8.91	9.07	9.23	9.39
60	9.55	9.71	9.87	10.03	10.19	10.35	10.50	10.66	10.82	10.98
70	11.14	11.30	11.46	11.62	11.78	11.94	12.10	12.25	12.41	12.57
80	12.73	12.89	13.05	13.21	13.37	13.53	13.69	13.85	14.01	14.16
90	14.32	14.48	14.64	14.80	14.96	15.12	15.28	15.44	15.60	15.76
100	15.92	16.07	16.23	16.39	16.55	16.71	16.87	17.03	17.19	17.35
110	17.51	17.67	17.83	17.98	18.14	18.30	18.46	18.62	18.78	18.94
120	19.10	19.26	19.42	19.58	19.73	19.89	20.05	20.21	20.37	20.53
130	20.69	20.85	21.01	21.17	21.33	21.49	21.65	21.80	21.96	22.12
140	22.28	22.44	22.60	22.76	22.92	23.08	23.24	23.40	23.55	23.71
150	23.87	24.03	24.19	24.35	24.51	24.67	24.83	24.99	25.15	25.31
160	25.46	25.62	25.78	25.94	26.10	26.26	26.42	26.58	26.74	26.90
170	27.06	27.21	27.37	27.53	27.69	27.85	28.01	28.17	28.33	28.49
180	28.65	28.81	28.97	29.13	29.28	29.44	29.60	29.76	29.92	30.08
190	30.24	30.40	30.56	30.72	30.88	31.04	31.19	31.35	31.51	31.67
200	31.83	31.99	32.15	32.31	32.47	32.63	32.79	32.95	33.10	33.26
210	33.42	33.58	33.74	33.90	34.06	34.22	34.38	34.54	34.70	34.85
220	35.01	35.17	35.33	35.49	35.65	35.81	35.97	36.13	36.29	36.45
230	36.61	36.76	36.92	37.08	37.24	37.40	37.56	37.72	37.88	38.04
240	38.20	38.36	38.51	38.67	38.83	38.99	39.15	39.31	39.47	39.63
250	39.79	39.95	40.11	40.27	40.42	40.58	40.74	40.90	41.06	41.22
260	41.38	41.54	41.70	41.86	42.02	42.18	42.34	42.49	42.65	42.81
270	42.97	43.13	43.29	43.45	43.61	43.77	43.93	44.09	44.25	44.40
280	44.56	44.72	44.88	45.04	45.20	45.36	45.52	45.68	45.84	46.00
290	46.15	46.31	46.47	46.63	46.79	46.95	47.11	47.27	47.43	47.59

Find pitch of teeth if $R = 20$ and $N = 93$. The constant for $93 = 14.8$ and $P = \frac{R}{C} = \frac{20}{14.8} = 1.35$ in.

Find number of teeth when $R = 32$ and $P = 2$ in.

We have $C = \frac{R}{P} = \frac{32}{2} = 16$, and reference to the table of constants shows that the nearest number corresponding to this is 16.07 at the intersection of 100 and 1 and therefore $N = 101$ teeth.

TABLE 33.—CENTRIFUGAL FORCE IN LB. OF A WEIGHT OF 1 LB. AT VARIOUS SPEEDS.

Dia. of Path	Revolutions per Minute.								
In.	100	200	300	400	500	600	700	800	900
12	1.70	6.81	15.32	27.24	42.56	61.3	83.43	108.97	137.93
18	2.55	10.22	22.98	40.86	63.85	91.95	125.16	163.47	206.90
24	3.41	13.63	30.65	54.49	85.13	122.60	166.87	217.95	275 86
30	4.26	17.03	38.31	68.56	106.42	153.26	208.60	272.42	344.80
36	5.11	20.44	45.96	81.73	127.70	183.90	250.32	326.94	413.80
42	5.96	23.84	53.63	95.35	149.00	214.55	292.00	381.40	482.70
48	6.81	27.25	61.29	108.97	170.26	245.20	333.74	435.90	551.72
54	7.66	30.66	68.96	122.60	191.57	275.88	375.45	490.40	620.65
60	8.51	34.06	76.61	137.12	212.84	306.52	417.20	544.84	689.60
66	9.36	37.47	84.28	149.85	234.13	337.16	458.90	599.40	758.70
72	10.21	40.88	91.94	163.45	255.39	367.80	500.61	653.85	837.58

Dia. of Path	Revolutions per Minute.								
In.	1000	1100	1200	1400	1600	1800	2000	2500	3000
12	170.25	206.00	245.15	333.75	435.90	551.65	681.00	1064.00	1532.25
18	255.40	309.0	367.7	500.6	653.8	827.4	1021.6	1596.2	2298.8
24	340.50	412.0	490.3	667.5	871.8	1103.3	1362.0	2128.0	3064.5
30	425.60	514.9	612.9	834.4	1089.8	1379.0	1543.6	2660.2	3831.05
36	510.80	618.0	735.5	1001.3	1307.6	1654.9	2043.0	3192.0	4596.7
42	595.90	721.0	858.0	1168.2	1525.8	1931.0	2383.4	3724.0	5362.8
48	681.00	824.0	980.6	1335.0	1743.6	2206.6	2724.0	4256.0	6129.0
54	766.20	927.0	1103.3	1502.2	1961.5	2482.8	3065.0	4789.0	6897.0
60	852.00	1039.9	1235.8	1768.8	2179.6	2758.0	3087.2	5320.4	7662.1
66	936.40	1133.2	1348.5	1835.8	2397.7	3034.2	3745.5	5852.5	8427.8
72	1021.50	1236.0	1470.9	2002.5	2615.2	3309.9	4086.0	6384.0	9193.5

TABLE 34.

SQUARES, CUBES, SQUARE ROOTS AND CUBE ROOTS.

No.	Square	Cube.	Square Root. $\sqrt{}$	Cube Root. $\sqrt[3]{}$	No.	Square	Cube.	Square Root. $\sqrt{}$	Cube Root $\sqrt[3]{}$
1	1	1	1·000	1·000	51	2601	132651	7·141	3·708
2	4	8	1·414	1·259	52	2704	140608	7·211	3·732
3	9	27	1·732	1·442	53	2809	148877	7·280	3·756
4	16	64	2·000	1·587	54	2916	157464	7·348	3·779
5	25	125	2·236	1·709	55	3025	166375	7·416	3·802
6	36	216	2·449	1·817	56	3136	175616	7·483	3·825
7	49	343	2·645	1·912	57	3249	185193	7·549	3·848
8	64	512	2·828	2·000	58	3364	195112	7·615	3·870
9	81	729	3·000	2·080	59	3481	205379	7·681	3·892
10	100	1000	3·162	2·154	60	3600	216000	7·745	3·914
11	121	1331	3·316	2·223	61	3721	226981	7·810	3·936
12	144	1728	3·464	2·289	62	3844	238328	7·874	3·957
13	169	2197	3·605	2·351	63	3969	250047	7·937	3·979
14	196	2744	3·741	2·410	64	4096	262144	8·000	4·000
15	225	3375	3·872	2·466	65	4225	274625	8·062	4·020
16	256	4096	4·000	2·519	66	4356	287496	8·124	4·041
17	289	4913	4·123	2·571	67	4489	300763	8·185	4·061
18	324	5832	4·242	2·620	68	4624	314432	8·246	4·081
19	361	6859	4·358	2·668	69	4761	328509	8·306	4·101
20	400	8000	4·472	2·714	70	4900	343000	8·366	4·121
21	441	9261	4·582	2·758	71	5041	357911	8·426	4·140
22	484	10648	4·690	2·802	72	5184	373248	8·485	4·160
23	529	12167	4·795	2·843	73	5329	389017	8·544	4·179
24	576	13824	4·898	2·884	74	5476	405224	8·602	4·198
25	625	15625	5·000	2·924	75	5625	421875	8·660	4·217
26	676	17576	5·099	2·962	76	5776	438976	8·717	4·235
27	729	19683	5·196	3·000	77	5929	456533	8·774	4·254
28	784	21952	5·291	3·036	78	6084	474552	8·831	4·272
29	841	24389	5·385	3·072	79	6241	493039	8·888	4·290
30	900	27000	5·477	3·107	80	6400	512000	8·944	4·308
31	961	29791	5·567	3·141	81	6561	531441	9·000	4·326
32	1024	32768	5·656	3·174	82	6724	551368	9·055	4·344
33	1089	35937	5·744	3·207	83	6889	571787	9·110	4·362
34	1156	39304	5·830	3·239	84	7056	592704	9·165	4·379
35	1225	42875	5·916	3·271	85	7225	614125	9·219	4·396
36	1296	46656	6·000	3·301	86	7396	636056	9·273	4·414
37	1369	50653	6·082	3·332	87	7569	658503	9·327	4·431
38	1444	54872	6·164	3·361	88	7744	681472	9·380	4·447
39	1521	59319	6·244	3·391	89	7921	704969	9·433	4·464
40	1600	64000	6·324	3·420	90	8100	729000	9·486	4·480
41	1681	68921	6·403	3·448	91	8281	753571	9·539	4·497
42	1764	74088	6·480	3·476	92	8464	778688	9·591	4·514
43	1849	79507	6·557	3·503	93	8649	804357	9·643	4·530
44	1936	85184	6·633	3·530	94	8836	830584	9·695	4·546
45	2025	91125	6·708	3·556	95	9025	857375	9·746	4·560
46	2116	97336	6·782	3·583	96	9216	884736	9·797	4·578
47	2209	103823	6·855	3·608	97	9409	912673	9·848	4·594
48	2304	110592	6·928	3·634	98	9604	941192	9·899	4·610
49	2401	117649	7·000	3·659	99	9801	970299	9·949	4·626
50	2500	125000	7·071	3·684	100	10000	1000000	10·000	4·641

TABLE 34.—CONTINUED.

SQUARES, CUBES, SQUARE ROOTS AND CUBE ROOTS.

No.	Square.	Cube.	Square Root. $\sqrt{}$	Cube Root. $\sqrt[3]{}$	No.	Square	Cube.	Square Root. $\sqrt{}$	Cube Root. $\sqrt[3]{}$
105	11025	1157625	10·24	4·717	360	129600	46656000	18·97	7·113
110	12100	1331000	10·48	4·791	365	133225	48627125	19·10	7·146
115	13225	1520875	10·72	4·862	370	136900	50653000	19·23	7·179
120	14400	1728000	10·95	4·932	375	140625	52734375	19·36	7 211
125	15625	1953125	11·18	5·000	380	144400	54872000	19·49	7·243
130	16900	2197000	11·40	5·065	385	148225	57066625	19·62	7 274
135	18225	2460375	11·61	5·129	390	152100	59319000	19·74	7·306
140	19600	2744000	11·83	5·192	395	156025	61629875	19·87	7·337
145	21025	3048625	12·04	5·253	400	160000	64000000	20·00	7 368
150	22500	3375000	12·24	5·313	405	164025	66430125	20·12	7·398
155	24025	3723875	12·44	5·371	410	168100	68921000	20·24	7·428
160	25600	4096000	12·64	5·428	415	172225	71473375	20·37	7·459
165	27225	4492125	12·84	5·484	420	176400	74088000	20·49	7·488
170	28900	4913000	13·03	5·539	425	180625	76765625	20·61	7·518
175	30625	5359375	13·22	5·593	430	184900	79507000	20·736	7·547
180	32400	5832000	13·41	5·646	435	189225	82312875	20·856	7·576
185	34225	6331625	13·60	5·698	440	193600	85184000	20·976	7·605
190	36100	6859000	13·78	5·748	445	198025	88121125	21·095	7·634
195	38025	7414875	13·96	5·798	450	202500	91125000	21·213	7·663
200	40000	8000000	14·14	5·848	455	207025	94196375	21·330	7·691
205	42025	8615125	14·31	5·896	460	211600	97336000	21·447	7·719
210	44100	9261000	14·49	5·943	465	216225	100544625	21·563	7·747
215	46225	9938375	14·66	5·990	470	220900	103823000	21·679	7·774
220	48400	10648000	14·83	6·036	475	225625	107171875	21·794	7·802
225	50625	11390625	15·00	6·082	480	230400	110592000	21·908	7·829
230	52900	12167000	15·16	6·126	485	235225	114084125	22·022	7·856
235	55225	12977875	15·32	6 171	490	240100	117649000	22·135	7 883
240	57600	13824000	15·49	6·214	495	245025	121287375	22·248	7·910
245	60025	14706125	15·65	6·257	500	250000	125000000	22·360	7 937
250	62500	1562 000	15·81	6·299	510	260100	132651000	22·583	7·989
255	65025	16581375	15·96	6 341	520	270400	140608000	22·803	8·041
260	67600	17576000	16·12	6·382	540	291600	157464000	23·237	8·143
265	70225	18609625	16·27	6·423	560	313600	175616000	23·664	8·247
270	72900	19683000	16·43	6·463	580	336400	195112000	24·083	8·339
275	75625	20796875	16·58	6·502	600	360000	216000000	24·494	8·434
280	78400	21952000	16·73	6·542	625	390625	244140625	25·000	8·549
285	81225	23149125	16·88	6·580	650	422500	274625000	25·495	8·662
290	84100	24389000	17·02	6·619	675	455625	307546875	25·980	8·772
295	87025	25672375	17·17	6·655	700	490000	3430 0000	26·457	8·879
300	90000	27000000	17 32	6·694	725	525625	381078125	26·925	8·983
305	93025	28372625	17·46	6·731	750	562500	421875000	27·386	9·085
310	96100	29791000	17 60	6·767	775	600625	465484375	27·838	9·185
315	99225	31255875	17·74	6·804	800	640000	512000000	28·284	9·283
320	102400	32768000	17·88	6·839	825	680625	561515·25	28·722	9·378
325	105625	34328125	18·02	6·875	850	722500	614125000	29·154	9·472
330	108900	35937000	18·16	6·910	875	765625	669921875	29·580	9 564
335	112225	37595375	18·30	6·945	900	810000	729000000	30·000	9·654
340	115 00	39304000	18·43	6·979	925	855525	7 1453125	30·413	9·743
345	119025	41063625	18·57	7·013	9 0	902500	857375000	30·822	9·830
350	122500	42875000	18·70	7·047	975	950625	926859375	31·224	9 915
355	126025	44738875	18·84	7·080	1000	1000000	1000000000	31·622	10·000

SYSTEMS OF LUBRICATION.

Chain or Ring System.—The main defect in this method of lubrication is that through neglect the oil is liable to become loaded with dirt and foreign matter to such an extent that it ceases to lubricate the bearings unless it is all drained out, and fresh oil put in every week or two. If this is done, the bearings may be fairly well lubricated by the use of this system.

A Circulating Oil System.—What is needed to secure a perfect system of lubrication is a continuous feed arrangement, either sight or forced, from a central reservoir, the oil being piped to each bearing in such a manner that the flow of oil ceases when the motion stops, and commences automatically again as soon as the machinery begins to move. The principal elements in a system of this kind are found to be oil pipes connecting each bearing with a reservoir of oil, to which flows all the oil from each bearing in the mill, but passing through a good oil filter while on the way from bearing to reservoir.

All the metal particles and other foreign matter are thus removed from the oil which is then raised from the reservoir by means of a small pump, a certain quantity being permitted to pass through sight-feed oil-cups into pipes leading to the several bearings in the mill. The surplus oil which is raised by the pump, and refused by the several sight-feeds, flows back again into the reservoir, from which it again passes through the pump as described. When the machinery stops, the pump stops also, and lubrication ceases as soon as the small quantity of oil adjacent to the sight-feeds flows back to the reservoir.

The oil being filtered perfectly clean, there is noth-

ing to stop up the sight-feeds and they require very little attention after having once been set for the amount of oil required by each bearing. As the amount thus sent to each bearing is always to be an excess of the quantity required, the sight-feeds are easily adjusted, and they remain in adjustment for a long time. This system is very easily applied to ring or chain oiling bearings, nothing being necessary except to connect the supply pipe through the usual oil-filling hole, then to tap in an overflow or return pipe by means of which the surplus oil is carried back to the reservoir.

To connect up rigid flat boxes, and that style of bearings, it is necessary to pipe the oil into the oil-hole, then add a drip pan under the bearing and connect each drip pan with a return pipe to the oil reservoir to carry back the surplus oil. Fast running bearings must have oil collars attached in order that the oil may not run along the shafts and sneak away past the added drip pans. But as most machines containing fast running shafts are made with the necessary oil collars, the problem in that direction is not a serious one.

Grease for Lubrication.—While grease is not altogether suitable for lubricating machinery running at high speed, it is well adapted for use on slow moving journals. For elevators, conveyors, and for all journals where there is more or less dirt grease is desirable for purposes of lubrication.

In case of ordinary grease lubrication, the grease should be contained in a screw grease-cup, and by screwing down the cover of the cup, a portion of the grease is forced through the regular oil passage into the bearing. It would seem at first sight as if this were a very poor method of lubricating, but upon closer observation there appears much to commend. The lubrication

is positive; the grease must go directly into the bearing as it can escape in no other way. An excess of grease will, if forced into the bearing, find its way out at the ends of the journal, and from rings or ridges around the shaft, effectually closing the openings to the entrance of sand or other dirt which may be adjacent to the bearing. Thus the grease-cup protects the bearing from sand in the case of elevators and conveyors.

A quantity of grease piled up at the ends of the bearing tells the oiler that too much grease is being used, and he will not force through as much on his periodical oiling trips. The worn-off metal is forced out of the bearing with the grease which is displaced daily by the fresh supply from the grease-cup, hence the bearings are kept clean and do not become filled up with worn-off shaft or babbitt metal. Should the shaft or the bearing become hot, the heat will be communicated to the grease in the screw-cup, and the expansion of the mass by heat will cause an extra portion of lubricant to flow into the bearing, thereby supplying the lacking lubricant and cooling the heated bearing to normal temperature. Grease lubricating also prevents the waste of oil so often seen where periodical or squirt-can lubrication is employed. The oiler cannot waste grease unless he does it on purpose, for he cannot pour one drop in the oil-hole and ten drops outside of the hole as is so often done when chasing the squirt-can. Taken all in all, grease lubrication is a good thing for slow-moving machinery, and the millwright is safe in using that method of lubrication on all bearings running less than 150 r.p.m.

Best Lubricants for Various Purposes.—According to tests made by Prof. Thurston, lubricants are listed as follows:

Low temperatures.—Light mineral lubricating oils.

Very great pressures, slow speed.—Graphite, soapstone and other solid lubricants.

Heavy pressures with slow speeds.—The above, and lard, tallow and other greases.

Heavy pressures and high speeds.—Sperm oil, castor oil and heavy mineral oils.

Light pressures and high speeds.—Sperm, refined petroleum, olive, rape, cotton-seed oils.

Ordinary machines.—Lard oil, tallow oil, heavy mineral oils and the heavier vegetable oils.

Steam cylinders.—Heavy mineral oils, lard, tallow.

Watches and other delicate machinery.—Clarified sperm, neat's foot, porpoise, olive, and light mineral oils.

For mixture with mineral oils, sperm is best, lard is much used, olive and cotton-seed are good.

Cold Tests of Oils.—One of the most important features in the selection of lubricating oils is the lowest temperature that the oil will stand, and still remain fluid. The determination of this property of oil is termed the cold test, performed as follows: A small quantity of the oil to be tested is placed in a glass jar, or large-mouthed bottle which is then packed in a mixture of ice and salt until the oil solidifies. The bottle is then removed from the cold pack, and the oil allowed to soften, being stirred constantly, while at the same time a thermometer is inserted, and the temperature at which the oil again runs is noted.

Efficiency of Different Systems of Lubrication.—Friction varies considerably with the method employed to lubricate bearings.

The friction-reducing efficiency of several different

modes of applying oil to bearings, is given in the following table:

EFFICIENCY OF DIFFERENT SYSTEMS OF LUBRICATION, AS SHOWN BY THE FRICTION OF THE JOURNALS OF A STEEL SHAFT REVOLVING IN BEARINGS OF GUN-METAL, LUBRICATED WITH THE SAME KIND OF OIL APPLIED IN DIFFERENT WAYS.

System of Lubrication.	Co-efficient of Friction.	System of Lubrication.	Co-efficient of Friction.
Forced lubrication	.003	Needle-lubricators	.023
Oil-bath-lubrication	.004	Intermittent, or ordinary oiling, average best results with	.025
Splash-lubrication	.005	Intermittent or ordinary oiling by hand with an oil-can, average results with.	.050
Saturated oil-pad lubrication	.014		
Self-lubricating bearings	.016		
Syphon oil-lubricators	.018		
Sight-feed lubricators	.020		

Efficient lubrication is best obtained by applying the oil through three oil-holes, one being placed at each side and another at the center of the bearing.

Lubricants for the Bearings of the Journals of Shafts. —The viscosity or body of an oil represents its fluidity, and is the measure of its resistance to being squeezed out of a bearing. The greater the viscosity the less is the liability to being squeezed out. The viscosity of oils varies considerably, but at a temperature of 65° Fahr. it is frequently as follows:

	Viscosity Compared with that of Sperm Oil.		Viscosity Compared with that of Sperm Oil.
Sperm Oil	1.00	Rape oil	2.30
Light mineral oil	1.70	Neatsfoot oil	2.40
Cotton-seed oil	1.90	Tallow oil	2.50
Olive oil	2.00	Heavy mineral oil	4.00
Lard oil	2.20	Castor oil	23.00

The viscosity decreases as the temperature increases up to 220° Fahr., when all these oils have nearly the same viscosity. This shows the importance of keeping bearings cool, especially those using oil of full body.

The best lubricants for the bearings of engines, shafting and machinery, under ordinary working conditions, are generally as follows:

Description of Bearings.	Description of Lubricant.
For high-speed shafts with a light pressure on the bearings.....	Mixed with Mineral Oil. Sperm oil.
For low-speed shafts with a light pressure on the bearings.....	Olive oil.
For high-speed shafts with a heavy pressure on the bearings.....	Sperm oil and castor oil.
For low-speed shafts with a heavy pressure on the bearings.....	Rape oil and lard oil.
For the crank-shafts of large steam-engines	Castor oil.

The nature of the bearing-metal is an important factor in economical lubrication. It was found in the case of a heavily loaded shaft that some bearing-metals required more oil than others to maintain the same cool temperature. The relative consumption of oil by different bearing-metals is in some cases as follows:

	Relative Consumption of Oil for an Equal Temperature.
Cast-iron bearings for the journals of shafts.....	1.00
Brass bearings for the journals of shafts..	.90
Gun-metal bearings for the journals of shafts.....	.85
Phosphor-bronze bearings for the journals of shafts.....	.80
Antifriction white-metal bearings for the journals of shafts....	.75

The consumption of oil by bearings of hardened cast-steel, for equality of temperature, is generally at least twice as great as that of gun-metal bearings.

Mineral Oil of fine quality is very effective in cleaning metal, and when used in combination with other oils as a lubricant, it tends to maintain bearings clean and free from clogging. Hence, mixtures of fatty oils and mineral oils are generally more efficient for the lubrication of the bearings of the journals of shafts than either of these oils alone. The following mixtures of oils form efficient lubricants for some bearings:

Description of Bearings.	Composition of Lubricants
For high-speed shafts with light pressure on the bearings.	Sperm oil 20 per cent., olive oil 30 per cent., and light mineral oil 50 per cent.
For low-speed shafts with light pressure on the bearings.	Olive oil 50 per cent., and light mineral oil 50 per cent.
For high-speed shafts with heavy pressure on the bearings.	Lard oil 35 per cent., sperm oil 25 per cent., and heavy mineral oil 40 per cent.
For low-speed shafts with heavy pressure on the bearings.	Rape oil 50 per cent., lard oil 25 per cent., and tallow oil 25 per cent
For moderately high-speed shafts with very heavy pressure on the bearings.	Sperm oil 35 per cent., rape oil 20 per cent., lard oil 25 per cent., tallow oil 15 per cent., and fine plumbago 5 per cent.
For low-speed shafts with very heavy pressure on the bearings.	Graphite-grease, composed of 25 per cent. of tallow, 30 per cent. of palm oil, 33 per cent. of mineral oil, and 12 per cent. of fine graphite or plumbago.

The addition of graphite to lubricants reduces friction, and diminishes the cost of lubrication. Graphite, grease and other thick lubricants are only suitable for the journals of shafts having considerable play in the bearings.

Oils of inferior quality are not economical, and they sometimes contain acids which injure some metals, especially those containing much zinc and lead.

Heating of Journals and Bearings.—The heating of a revolving journal is produced by the friction of its surface on that of its bearing.

The intensity of the pressure on a bearing surface decreases as the length of a journal increases; and the heating tendency decreases as the area carrying off the heat is increased.

ERECTING STEAM ENGINES.

The setting of a steam engine should commence with the building of its foundation. The design of the foundation should be broad enough at the bottom, where it bears upon the soil, to ensure that no tremble or shake will ever cause the foundation to settle in whole or in part. The office of an engine foundation is threefold,

or in other words it must fill three requirements, either fully or partially, according to the surroundings. Briefly

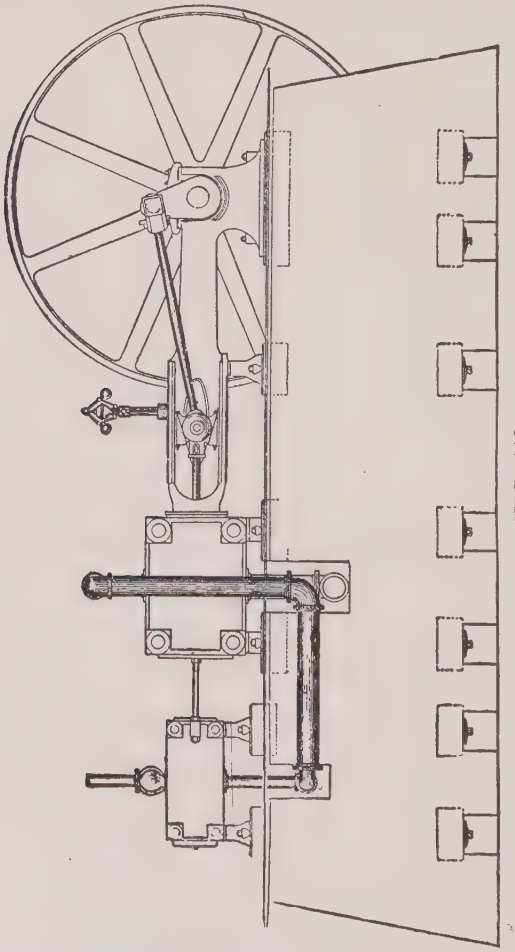


FIG. 112.

stated, these three functions of the engine foundation are:

I.—To support the mass of the engine at a given height or level.

II.—To preserve alignment of engine with the main shaft.

III.—To absorb vibrations of the moving parts of the engine.

Construction of Engine Foundations.—The foundation is to absorb the vibrations of the engine. It is therefore necessary that it be constructed in a single mass, or at least be so connected that one portion cannot move independent of the other portions. For this reason it is best to build the foundation of concrete in a single block if possible. A good, well built concrete foundation is preferable to any other kind, although brick or stone may be used to advantage if these materials are cheapest. Where the importance of the structure warrants it, small tunnels, or thoroughfares, should be made or left through the mass of the foundation by means of which access may be had to the lower ends of the bolts by which the bed-plate is bolted to the masonry (see Fig. 112).

For small engines, footings of rough masonry or ashlar may be used to distribute the pressure, and on this footing the foundation proper of brick may be built. The third plan is to make the foundation a monolith of concrete. Upon a proper footing to distribute the weight, a box of rough boards without top or bottom is laid, and within it successive layers of cement concrete are thrown in and well rammed until the desired height is reached. When brick is used it should be of first quality, hand-burned, and laid in cement-mortar. Common lime-mortar is liable to crumble and disintegrate under vibration, and the whole principle of the foundation is to have it act as a solid mass. When ap-

pearance is to be considered, the face of the brick foundation may be made of face, or Philadelphia brick, while the interior is of ordinary grades. Since the bed-plate is to be bolted to the foundation, the greatest care must be observed in locating the necessary bolts in their proper places.

Footings to Prevent Vibration.—The mass which it is convenient to get in a vertical engine bed-plate is often not enough to provide for the absorption of all vibration. The vertical engine, when chosen because floor-space is to be saved, does not call for extended area in the foundation, so that sufficient mass can only be gotten by going deep. Where this is inconvenient, or where rock is struck, engineers have had to provide special footings to arrest vibration. It has been tried by some to underlay the foundation proper with timber or rubber, but a springing material of a class to which these belong often causes the very difficulties they are designed to prevent.

Vibration of machinery, or any solid substance is of two sorts: the material either swings crosswise as in the vibrating string of a musical instrument or in a flapping belt, or the motion of the particles is lengthwise or parallel to their long axis. If the oscillation, or vibrating period of the material used as an absorber of vibration happens to coincide with the vibration period caused in the engine frame by the speed of reciprocation, or by the belt flap, the deadener partakes, and multiplies the objectionable vibration. Probably no better material is to be found for the purpose of stopping vibration than sand, provided it can be kept dry, and all motion prevented, and the foundation-block itself is of sufficient mass. The foundation pit should be dug two or three feet deeper, and two or three feet

wider than the foundation proper is to be. This pit is surrounded with the proper sheathing to prevent the displacement of the sand, which is filled in two or three feet below the bottom of the foundation, and then around it on the sides as it is built up. Hair-felt or mineral wool layers have been used underneath the footing-course. But if the foundation-block is not massive enough, these methods only serve to aggravate the difficulty which they are intended to cure. Very satisfactory results have been obtained abroad from the use of asphaltic concrete for massive footings.

This material possesses a certain sort of elasticity with its massive character, and its period of vibration is so definite, and so much shorter than the period of the engine's vibrations, that the latter are broken up and neutralized before they reach the transmitting rock, or hard-pan. Most annoying vibrations are often caused in high-speed engines by the impact of steam in an exhaust-pipe with elbows. The difficulty is intensified when there are water, or oil drops in the exhaust current. Their impact against the elbow which deflects them will set lengths of pipe a-tremble, and their motion will be transmitted over a very extensive area.

Foundation Bolts.—The bed-plate of the engine requires to be strongly and rigidly secured to the foundation in order that the latter may act with the bed-plate as one mass, and thus prevent the latter from moving on the foundation.

These anchor bolts vary in size with the size of the engine, but they should never be of such small diameter that it can be possible to twist them off with an ordinary wrench. Common diameters of anchor bolts for engines of medium size are from 1⅛ to 1½ inches.

The larger sizes of engines require 2-inch bolts, while

for the smallest sizes, ¾-inch bolts are sufficiently strong, if properly handled in screwing down the nuts. Regarding the lengths of anchor bolts, it is best to have them extend all the way to the bottom of the foundation, if possible, in order that the upward strain upon them may be widely distributed in the foundation.

The location of these bolts in the foundation is determined by the holes in the bed-plate through which they are to pass. The typical bed-plate as a general

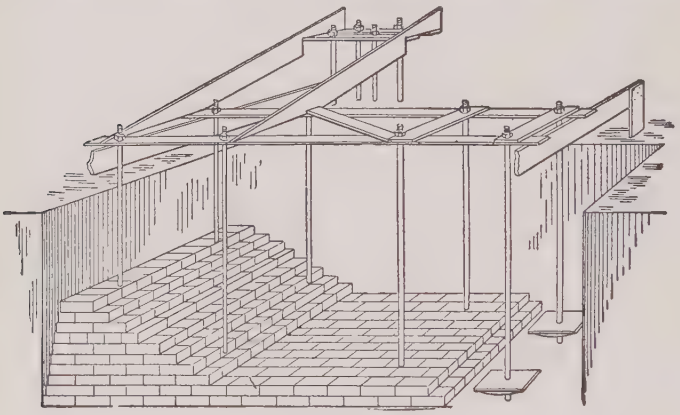


FIG. 113.

rule has bolt holes at the cylinder end, in the feet, and at the crank shaft (see Fig. 112). The bolts, furthermore, must be built into the foundation, and be of such length that when the foundation is completed, and the bed-plate placed upon it, the upper ends of the bolts will protrude through the holes in the bed-plate a sufficient distance to receive the nuts which these upper ends are to carry.

The method used to accomplish this is illustrated in Fig. 113, which presents a typical arrangement for this

purpose. The wooden frame is called a template. It has holes made through it at points which correspond to the holes in the bed-plate, and when the nuts are in place on the upper end of the bolts the template is adjusted to the proper height above the datum plane, or plane of reference, and the foundation is built around the hanging bolts. The lower ends of the bolts are fitted with thread and nuts, on top of which rest the bearing or distributing plates or washers of cast or wrought iron. The distributing-plate is to enable the effort of the bolt to be borne by a number of bricks without danger of pulling through, and the nut and thread permit a vertical adjustment of the bearing-plate so that it shall come at the under surface of a joint in the coursed masonry. To permit of a certain limited horizontal adjustment of these foundation-bolts, each bolt should be surrounded with a length of pipe reaching from the bearing-plate to the top of the masonry. The diameter of this pipe is so chosen that the bolt can be deflected within the hole which the pipe makes, and after the bolt is in place and the alignment completed, the space between the bolt and the sides of the pipe is filled with concrete, and the position of the bolt is fixed. The template in Fig. 113 shows the anchor bolts for the outboard pillow-block attached to the principal template. This is usual when drawings of the template are furnished by the engine builder, and it is desired to make the foundation all in one block.

In cases where the length of the engine shaft makes it desirable to have a separate foundation for the outer bearing, it is usually more convenient to work with an independent template.

Constructing a template.—A template is usually made of $\frac{7}{8}$-inch boards fastened together with clinch nails

or screws. Fig. 114 shows a template layout as arranged for an engine of the detached outboard pillow-block type.

Center line a, b, sketch A, is first drawn on a smooth floor or platform large enough to permit laying out of the foundation full size.

Shaft line b, c, is next drawn exactly at right angles to line a, b. Next lay off the distance b, d, equaling the

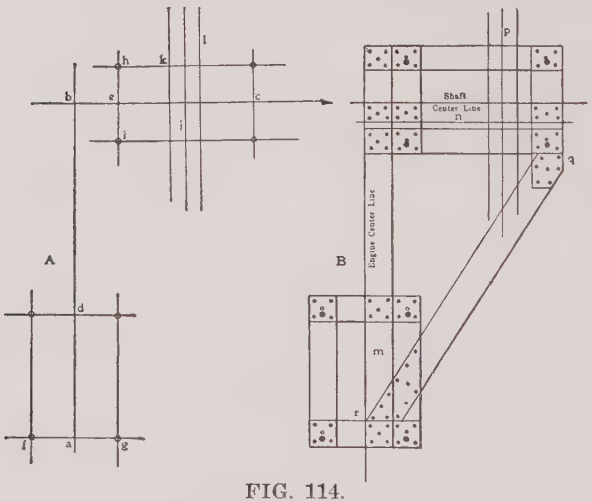


FIG. 114.
CONSTRUCTING A TEMPLET.

space between the shaft line and the anchor bolt holes under the front end of the cylinder.

Next find distance a, d, and lay it down, locating the holes under the head end of the cylinder. The distances a, f, and a, g, locate the spaces between the centers of the holes at f and g at a, and at d.

Next locate the holes at h and i, on line e, also on line c, the latter being for the outboard pillow-block anchor bolts.

If there are to be other bolts in the foundation, they should be located as above. Some engines have an anchored arm or bracket to carry the end of the rock shaft. When such a bracket is attached to the foundation, it should be located on the diagram. When dashpots are used, they also should be located, and it is well to mark the diameter of the exhaust pipe where it drops below the floor level and a pocket or cavity must be left for it in the concrete. These things are not shown on the diagrams either at A or at B. The millwright may mark or omit them as he chooses. One thing should always be marked, and that is, the center line and faces of the engine pulley, as at j, which indicates the center line, while k and l represent the faces or sides of the pulley. These lines are very convenient for locating the template when it is necessary to bring the engine to a pulley already located upon the main shaft.

The diagram is now ready for the template which is to be built upon it. Procure some good sound lumber, white pine, spruce or yellow pine. Do not use whitewood, it warps too much. White pine is the best, but that is probably out of the question. Spruce will answer very well, and yellow pine makes a fine template but a very heavy one, as green yellow pine boards 1-inch thick weigh five pounds to the square foot. Use dry lumber. If necessary, dry it before making the template, for if green the template will not remain accurate. Hemlock may be used, but it is not very desirable lumber.

Have the boards planed on four sides if possible, on one side and one edge, anyway. It is possible to make a template from rough undressed boards, but it is not a desirable thing to do and the accuracy is apt to suffer

by so doing. Even with rough lumber, one edge of two boards must be jointed straight and placed one upon each center line as at m and n, sketch B, Fig. 114.

Some mechanics construct templates with the center line drawn down the middle of one of the boards, but a better plan is to use the straight-edge of the board as the center line whenever possible, as there is no danger of the line being rubbed out, which is likely to happen when drawn on the board. Sketch B, Fig. 114, shows plainly the method of constructing the template. The brace o should be carefully located and secured, in order that there be no shifting of the outboard pillow-block bolt holes. It should be noted that the brace, o, bears, or should bear firmly against the template at q and at r, the brace being carefully fitted at those points, then firmly clinch nailed or screwed as indicated in sketch B. The wheel location is shown at p, thus enabling the template to be readily brought to the correct point, and permitting the masons to see where the wheel is located, there enabling them to keep the foundation free from any possible interference with the lower rim of the wheel.

Setting up a Template.—The template may be placed on ledgers supported by posts like ordinary batters, or the ledgers may be suspended from overhead. This latter method is to be preferred, as it leaves the space around the foundation free from all timber or posts, which would interfere to some extent with the movement of workmen, and material. It is good practice to locate the under side of the template upon the finish level of the top of the foundation, thus allowing measurements to be taken from that surface.

Alignment of the Template.—The foundation, or anchor bolts bear a certain relation to the axis of the

cylinder. The axis, or center line of the cylinder must be in a plane truly at right angles with the center line of the crank shaft. If the engine shaft is to drive a line shaft by belting, or gearing, these two shafts must be exactly parallel. Hence it is of prime importance to have the cylinder axis perpendicular to the line of shafting, and the template which carries the anchor bolts must be very carefully placed or oriented with respect to these determining lines.

The drawing furnished by the builder of the engine from which the template is to be made usually has on it the center-line of the cylinder, so that it can be laid out upon the boards of the template.

For the obtaining of the vertical plane through the cylinder-axis a line stretched over the foundation-pit and carried to suspended plumb-bobs is the usual device. For laying off the center-line of shafting or wall-lines the expedient of snapping a chalk-line upon the floor is the most convenient. The centers of the shaft are transferred to the floor by plumb-lines, or offsets may be taken from permanent walls. Such center-lines having been established, the plane at right angles to it is established by points and lines using either a transit with graduated horizontal limb and making repeated readings, or by the ordinary geometric methods, or by the use of a massive T square whose head and blade exceed six feet in length and whose squareness has been carefully verified.

If a pulley or belt-wheel has been placed upon the line-shaft to which it is desired to draw a perpendicular line, a most convenient method is to stretch a twine or fine wire across the diameter of the pulley as nearly as the shaft will permit. With pulleys which have been turned, the edges of the face determine a plane perpen-

dicular to the axis, so that the tense string touching the face at one point will only touch the face at a point on the other side of the shaft when the further end of the string lies in a plane which is perpendicular to the axis. This same method is a very convenient one to extend for the purpose of bringing two shafts parallel to each other where both carry pulleys.

Locating the Bed-plate on the Foundation.—The foundation being completed, the bed-plate is to be lifted upon it and dropped into place with the bolts passing up through the holes in the bed-plate. Where cranes or similar lifting appliances are a feature of the power-house equipment this process becomes simple. In their absence the bed-plate must be lifted by jacks and blocking high enough to clear the bolts. It must then be rolled on skids into place by the successive withdrawing of the blocking.

The masons or bricklayers who have built the foundation do not usually have appliances for working to as close dimensions or as accurate levels as the setting of the engine requires. Furthermore, the top of the foundation is rarely a true plane, while the bottom of the bed-plate is very nearly a plane as a rule. It is necessary, therefore, to make a joint between the bed-plate and the masonry work which shall support the bed-plate all over and in a plane as nearly level as it can be made. This process is so much easier when the brick-work or jointed masonry is covered by a single flat cap-stone, that where the dimensions of the foundation permit its use is to be preferred. It is usually a sawed or planed slab of bluestone or flagstone from four to six inches thick, and a little larger than the foundation pier to which it serves as a finish or coping. The holes for the foundation bolts must be drilled in it, and

it is lowered to its place upon a good bedding of cement. In the absence of such a cap or coping the bearing of the bed-plate comes upon a surface which is full of joints. The bed-plate is lowered over the foundation bolts, and rests upon thin flat shims, or wedges of metal, which are placed on each side of the bolts between the bed-plate and the foundation. The nuts of the bolts are then screwed home, compressing the shims, while the bed-plate is carefully levelled as the strain is taken at each bolt. By driving in or loosening the shims any distortion or warping of the bed-plate by the bolts is prevented, and the bolts are tightened home until they refuse to go further.

The bed-plate is now rigidly bolted to the foundation and rests upon a number of points in a plane. Between the bed-plate and the foundation is a space between the shims equal to their thickness, and this joint requires to be filled. The materials used for this purpose in setting a bed-plate and making the joint are five. They are methods applicable to the setting of any machinery.

1. Shredded oakum may be driven into the joint with a chisel, as the seams of wooden vessels formerly were calked. This makes an elastic joint, but it lacks permanency.

2. Felted hair is used in the same way and has the same properties.

3. A rust-joint, as it is called, may be used. This is made by taking a thin cement grout into which cast iron borings or chips are introduced with a little powdered sal ammoniac and flour of sulphur. A dam of putty or clay is made around the outside of the bed-plate, and this mixture run into the joint and well

worked in with a trowel. The rusting metal unites the mixture to the iron, and the cement to the stone.

4. The sulphur-joint. This is one of the most widely used methods for bedding the engine. A clay or putty dam is made around the bed-plate, and the ordinary roll sulphur melted in an old kettle and poured into the joint between the bed and the masonry. It expands on solidification somewhat like ice to fill every interstice and give full support to the bed-plate. It undergoes no deterioration from oil or vibration. If care is not taken in melting the sulphur, it will be-

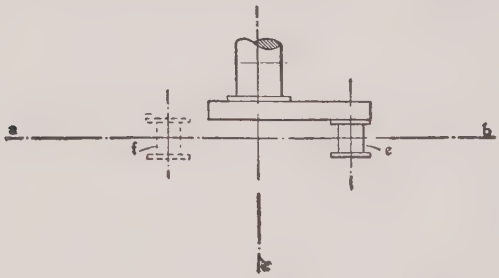


FIG. 115.
CHECKING ALIGNMENT OF ENGINE SHAFT.

come too hot and begin to oxidize, giving off an offensive gas. Sulphur in melting becomes fluid at a comparatively low temperature, becomes more viscid as the temperature rises, and passes to a second fluidity just before it is ready to burn.

5. The type-metal joint. Advantage is taken of the property possessed by certain antimony alloys (such as Babbitt, type metal, etc.) of expanding at the moment of solidification to use them for bedding or jointing bed-plates. The method of using them is the same as that practiced with sulphur, and they are preferred by

many engineers particularly for bedding the narrow feet used with Corliss bed-plates.

Alignment of Outer Pillow-Block or Shaft-Bearing. —Many engines have a bearing on each side of the crank pin and connecting rod. Such engines are called center-crank engines. The two bearings for the shaft are on the bed-plate, and the fly-wheels overhang their bearings. Side-crank engines, on the other hand, have a shaft bearing behind one crank on the bed-plate, but the outer end of the shaft requires an independent bearing. The fly-wheel or belt-wheel or both will usually be upon the length of the shaft between these two bearings.

It will be seen at once that the location of this outer bearing is of vital importance. In small engines it can be provided for approximately by the template as shown in Fig. 113, but for large engines and for its final adjustments in small engines a different method should be used.

If the outer bearing is too high or too low, it will force the crank to revolve in a plane making an angle with the true vertical plane, and twist the connecting-rod in each stroke. If out of place in a horizontal plane while correctly located in a vertical plane, it will force the crank to revolve in a plane which makes an angle with the axis of the cylinder, in which case it will bend the connecting rod in each stroke; or it may be out of place in both planes, so that the connecting rod will be both twisted and bent. The effect of either or both errors of alignment of this outer bearing is to wear the crank-pin out of its cylindrical shape, and to cause a knock or pound, and heating at the joint, which no adjustment of these bearings will cure. The proper method of aligning the outer bearing involves, first, the

establishment of the true axis of the cylinder after the bed-plate is in place and the foundation-joint complete. This is best done by stretching a fine piano-wire through the empty cylinder, carefully adjusting it to the center of the bore and fastening it tightly stretched to walls or fixed objects. To get the wire central is a matter of painstaking care and trial with gauges of wood or metal whose length is the cylinder radius. The axis of the cylinder being established, the shaft and crank are put in place in the bearing on the bed while the outer bearing is provisionally supported and located. The shaft is then turned until the crank coming towards its inner

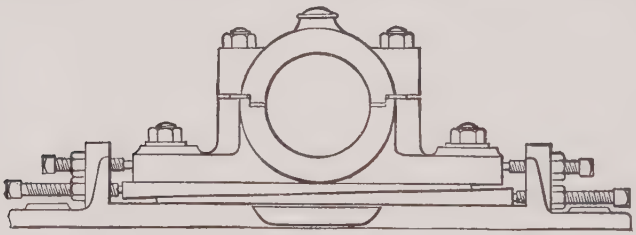


FIG. 116.

dead center touches the wire which marks the prolongation of the cylinder axis. It will touch it at a certain distance from the end of the crank pin, and from one of its collars.

Fig. 115 will serve to more clearly illustrate this method of alignment. The line drawn through the center of the cylinder is represented at a, b, while the center of the engine shaft is outlined at c, d. The crank being under the line a, b, the shaft is revolved until the crank-pin is brought as close as possible to the line without touching. While in this position the distance to the fixed collars on the pin is measured with a steel scale, or with inside calipers. The line b should be

equidistant between the fixed collars on the crank pin, as shown at e. After getting this measurement, the engine shaft is again revolved until the crank-pin again comes almost to the line at f, where the distance from the line to each collar is again carefully measured, and if found exactly equal to the measurements taken at e, the shaft may be considered "square" with line a, b, and also square with the cylinder. If the measurements at e and f do not agree, the outer bearing should be

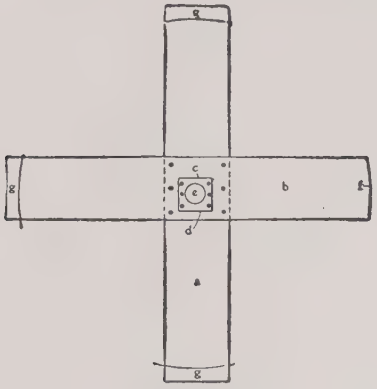


FIG. 117.
CROSS FOR CENTERING HEAD END OF CYLINDER.

adjusted horizontally by means of the set-screws shown in Fig. 116, until the wire cuts the crank-pin at the same point, and in the same plane on both its outer and inner centers. Reference has already been made in a general way to the method of centering the line passing through the engine cylinder.

A wooden cross (see Fig. 117) is made as follows: Two pieces of board 2½ or 3 inches wide, are halved together and screwed or clinch-nailed as shown; then a hole ⅞ inch or 1 inch in diameter is bored through at

c, a bit of tin or other thin metal, d, is tacked over the hole, and a small hole, e, just the size of the line used, is pierced through the tin as shown. This hole is used as a center from which to strike the circle f g g g, which is the exact diameter of the cylinder counterbore. A mark is made on each end of the wooden cross, as at g, g, then each arm is cut off exactly at the marks, as shown at f, thus leaving the small hole, e, in the exact center of the wooden cross which in turn fits snugly into the cylinder counterbore.

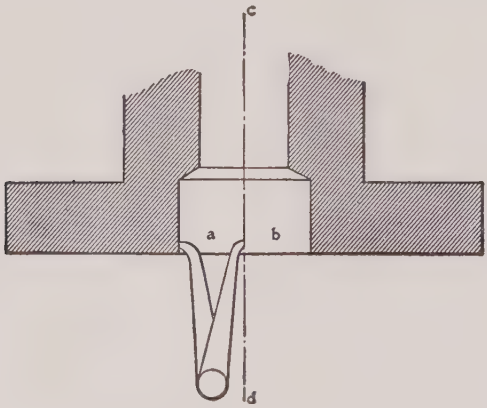


FIG. 118.
CENTERING LINE IN GLAND.

Adjusting the Center Line.—The line, c d, Fig. 118, consisting of a string or a wire, has been passed through hole e, Fig. 117, knotted, then threaded through the cylinder, from which it passes out by means of the rod hole and gland, as shown by Fig. 118, the free end of the line being attached to a target and pulled tight. The problem now becomes twofold. The line must be centered in the gland of the stuffing box by means of a pair of inside calipers which are applied as shown at

a, and also applied at b, the position of the engine bed being adjusted until the line c d lies as close to the center of the gland as human skill can determine. This is one part of the problem. The other is to maintain the line in its position in the center of the cylinder, and at the same time so move the engine bed that the line a b shall be brought perpendicular to, or square with the main shaft line.

Care should be taken, both in setting up the engine, and also in adjusting the main bearings, to see that the engine shaft is square, as above described, and that it remains so.

Where adjustment is required the usual procedure is followed of correcting half the error and testing the alignment again. The outer pillow-block is often made to rest upon a special foundation-plate which has provision for the adjustment upon it of the bearing proper in the horizontal plane (Fig. 116). The vertical adjustment otherwise is made either by shimming or filling in with sulphur or type metal below the plate, and the last and finest adjustment can be made by liners underneath the bearing-brasses. The alignment of vertical engines is usually simpler than that of horizontal engines, because the bearings are always on the bedplate and have been made right as to alignment by the builders in their shop-handling. The alignment in the erection of beam-engines is a simple and obvious extension of the principles laid down above. The vertical cylinder-axis, and the vertical line through the center of the crank pin when the latter is at the top, and at the bottom of its travel, determine the vertical plane in which the beam must play, and the crank pin at its 90 degrees and 270 degrees point must remain in that same plane.

The adjustment of the horizontal plane may be ef-

fected by a sensitive level if the engine has also been leveled in the plane at right angles to the cylinder-axis in setting upon the foundation. A more sensitive and satisfactory vertical adjustment of the out-board bearing may be accomplished by placing the crank pin at 90 degrees from its dead center, and holding a plumb line so as to touch the wire at the pin, then noting the distance of the vertical plane thus established, from the end of the pin, or collar. If the plumb line touching the wire also touches the crank pin at the same distance from the reference mark when the pin is at half-stroke below the wire, then the pin is turning in a vertical plane through the wire, and the outer bearing requires no vertical adjustment.

Equalizing the Clearance.—The clearance at either end of the cylinder should be watched very closely while adjusting the main bearings. Not only should the clearance be equalized, but it should be kept equal, not only when adjusting the bearings, but when taking them up against ordinary wear. Place the main bearing, also the outboard bearing, in the center of the housing; then test the clearance. This may be done in three ways, the first of which is to procure a bit of mirror—a pocket mirror will do and may be kept for the purpose—and prepare it by removing the silvering from a spot in the center of the glass about ¼ inch in diameter.

Then place a light close to one end of the cylinder, remove the plugs from the indicator holes, and with the glass in front of the eye and looking through the hole in the mirror, throw a beam of light into the indicator pipe hole, and observe the distance between the piston and the cylinder head when the crank is on the dead center. Make a similar observation at the other end of the cylinder. Then if the two distances do not

correspond, they should be equalized by adjusting the quarter boxes in the main bearings until the distances between the piston and cylinder heads, at both ends of the stroke are exactly equal to each other.

Another method of equalizing the clearance is as follows: Insert in each indicator pipe hole, or in the cylinder drip holes if there are no indicator openings, a piece of soft lead wire or small lead pipe as large as will go into the openings in question. Turn the engine pulley until the crank has been on the center in either direction, then withdraw the pieces of lead and note if they have both been flatted to the same thickness. If they have not, move the shaft in the proper direction one-half the difference of thickness of the two pieces of lead.

Care must be taken in flattening the pieces of lead not to allow them to project too far into the cylinder, otherwise the lead may be flatted out so wide that it could not be withdrawn through the indicator pipe hole. This would happen were the piston too close to one end of the cylinder. A little care in this direction will prevent fastening the lead in the holes. Sometimes when lead is not handy, a soft wood stick may be used instead. The flatting of the wood must be quickly noted, lest it springs back and leads the observer to believe that more room is present than really exists.

Another easily applied method is to place the engine on the dead center, and test the distance of the piston from the cylinder head by inserting a thin wooden wedge through the hole for the indicator pipe, then withdraw the wedge, and caliper its thickness at the point where it touched both the piston and cylinder head. This will show at a glance the clearance at that end of the stroke, then by placing the engine on the opposite center

and again inserting the wedge, the clearance at the other end may be ascertained and comparisons and adjustments made.

BOILER SETTING.

It generally falls to the lot of the millwright to superintend the setting of the boilers, and in many cases the installation of the piping also. Therefore a space should be devoted to a discussion of these two very important topics.

Location.—Boiler location is one of the first things which should be decided upon when a plant is projected. It should proceed closely with the location of the engine and the other machinery in the mill. In fact, the location of the power plant is one of the most vexatious problems of factory design. A balance must be struck between the cost of generating steam and the transmission of power; the matter of other steam consumption becomes a factor, such as heating the buildings, drying, boiling, and other chemical and mechanical operations.

The efficiency of a power plant depends a good deal upon the boiler, not only upon its efficiency as a steam generator, but upon the location and manner in which it is erected. A boiler may be so set that a great deal of heat is lost in the setting through air inlets, cramped passages and poor heat circulation, if that term may be allowed. In addition, there may be serious losses due to improper location of the boilers. It may cost too much to convey steam or power to the points of consumption, or perhaps the method of getting fuel to the boilers is a costly one. Again, the matter of removing ashes is difficult, or the boiler may be so located that repairs become a serious matter, even aside from the cost of similar repairs to other boilers. And when new

boilers must be put in, the cost of removing the old ones and substituting new ones becomes as great or even greater than the original cost of the boilers when new.

Setting.—In the following remarks concerning boiler setting, reference is had chiefly to the horizontal tubular boiler. Owing to the many and varied styles of water-tube boilers no prescribed set of rules is applicable, each builder of this type of boiler having a set plan of his own for the brick work. In the case of internally fired boilers the matter of setting resolves itself into the simple point of securing a sufficiently solid foundation, either of stone or brick laid in cement, for the boiler to rest upon. Concrete makes the

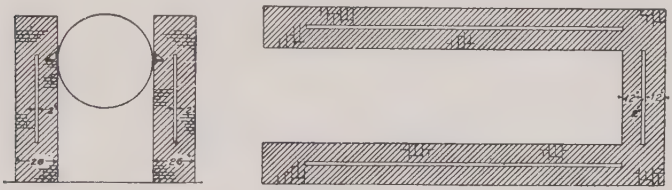


FIG. 119.

best foundation for steam boilers especially in cases where the nature of the soil is spongy or soft. In the case of installations where a number of boilers are to be set in battery, and the columns of the building are placed among the boiler foundations, it is necessary to place heavy, and absolutely safe foundations for the boiler walls in order to prevent settling. Regular setting plans should be drawn, and followed in setting boilers. The boiler setting should in fact be laid out and squared with the building line, in the same manner as with the other machinery.

Having laid out and staked off the ground on which the boilers are to stand, the next step is to roll the boilers

in place, jack them up a sufficient height, and build under and near each end a supporting crib-work of blocks 3 or 4 feet long by 5 or 6 inches square. Care should be taken in placing the boilers upon their temporary foundations of blocks, to see that they are exactly in the places they are to occupy permanently, that is, so far as vertical and horizontal distances are concerned.

Horizontal tubular boilers should always be set at least one inch lower at the back end than at the front, to

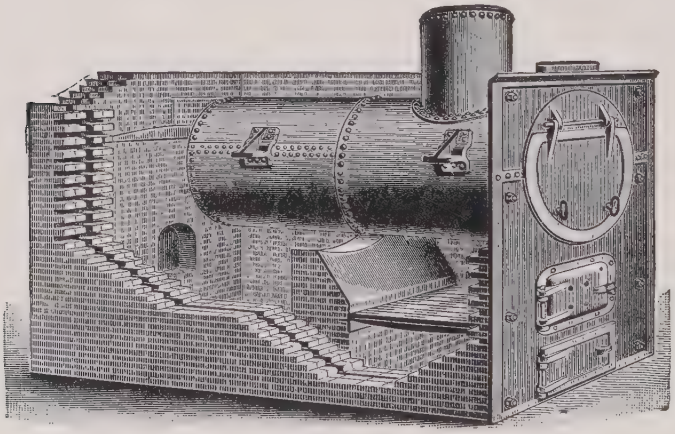


FIG. 120.
RETURN TUBULAR BOILER—SHOWING SETTING.

make sure that the rear ends of the tubes will be covered with water so long as any appears in the gauge glass, provided of course that the lower end of the glass is properly located with reference to the top row of tubes.

Usually it is not necessary to excavate very deep for a boiler foundation. A trench a foot or so in depth, following the outline of the walls and across where the bridge wall is to go, will be sufficient in most cases. The side walls, including the air space, are about 30

inches, and the rear wall, 28 inches. The bridge wall is about the same thickness, and the front wall may be 9 inches for a half arch front, or 16 to 18 inches for a full flush front boiler.

Buck Stays and Tie Rods.—After the concrete, or stone foundation has been laid, and before the brick work is started, the buck stays and lower tie rods should be located, and the latter placed in position. Boiler walls should always be well secured in both directions by tie rods extending throughout the entire length

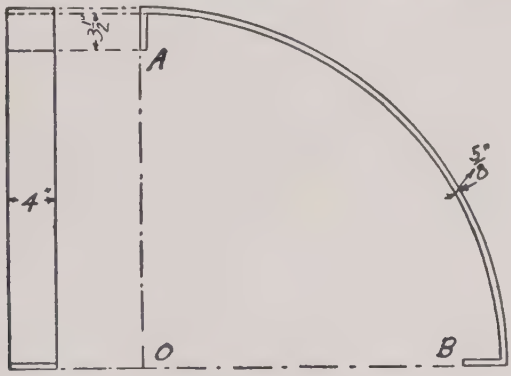


FIG. 121.

and breadth of the setting, whether there be one boiler or a battery of several. The bottom rods should be laid in place at the floor level when starting the brick work, and the top rods extending transversely across the boilers can be laid on top of the boilers. The top rods extending from front to back can be laid in the side walls or rest on top of them. All tie rods should be at least one inch in diameter, and for batteries of several boilers they should be larger. The rods should extend three or four inches beyond the brick work, with good threads and nuts on each end to receive the buck stays.

In laying down the transverse tie rods they should be located so as to allow the buck stays to bind the brick work where the greatest concentration of heat occurs.

In the case of the horizontal tubular boiler, there are two methods of support, one by suspension from I-beams and girders, to be described later on; the other by supporting the boiler upon brackets or lugs, riveted to the side sheets, and resting upon the side walls, and in such settings particular attention should be paid to securing a good foundation for the walls and great care exercised in building them in such manner that the expansion of the inner wall or lining will not seriously affect the outer walls. This can be done by leaving an air space of two inches in the rear and side walls, beginning at or near the level of the grate bars and extending as high as the fire line, or about the center line of the boiler. Above this height the wall should be solid. Fig. 119 shows a plan and an end elevation illustrating this idea. The ends of some of the bricks should be allowed to project at intervals from the outer walls across the air space, so as to come in touch with the inner walls.

Where boilers are set in batteries of two or more the middle or party walls should be built up solid from the foundation. All parts of the walls with which the fire comes in contact should be lined with fire brick, every fifth course being a header to tie the lining to the main wall.

Bridge Walls.—The boiler setting shown in Fig. 120 shows the bridge wall as being curved on top to conform to the circle of the boiler shell. Concerning this problem, much difference of opinion exists among engineers. The author's experience of many years has confirmed him in the belief that bridge walls should be built straight across from wall to wall of the setting, and

should not be curved to conform to the circle of the boiler shell. The proper distance from the top of the bridge wall to the bottom of the boiler varies from eight to ten inches, depending upon the size of the boiler. The space back of the bridge wall, called the combustion chamber, can be filled in with earth or sand, and should slope gradually downward from the back of the bridge wall to the floor level at the rear wall, and should be paved with hard burned brick. The ashes and soot can then be easily cleaned out by means of a long-handled

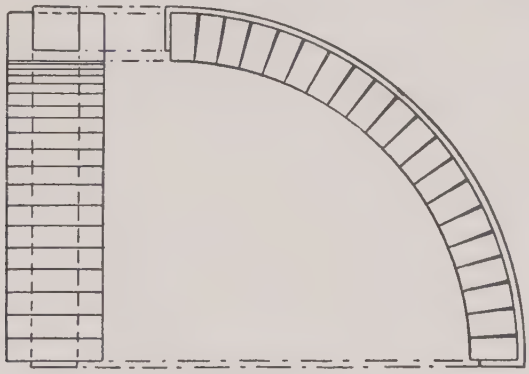


FIG. 122.

hoe or scraper inserted through the cleaning out door, which should always be placed in the back wall of every boiler setting.

Bearings for Lugs.—Upon the brick work and immediately under each lug of the boiler there should be laid in mortar a wrought or cast iron plate several inches larger in dimension than the bearing surface of the lug and not less than one inch in thickness. Upon each of these plates there should be placed two rollers made of round iron 1 or 1⅛ in. in diameter, and as long

as the width of the lug. These rollers should be placed at right angles to the length of the boiler, in such a position that the lug will bear equally upon them. The object of the rollers is to prevent disturbance of the brick work by the endwise expansion and contraction of the boiler.

Grate Surface.—The number of square feet of grate surface required depends upon the size of the boiler. A good rule and one easy to remember is to make the length

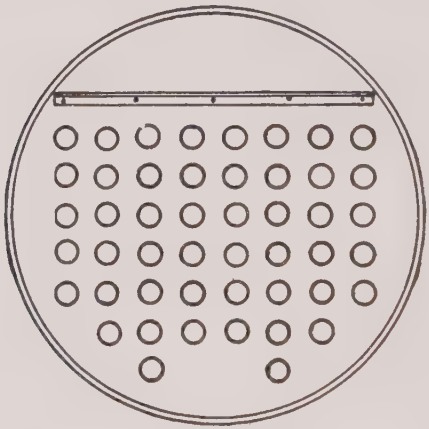


FIG. 123.

of the grates equal to the diameter of the boiler. The width, of course, will depend upon the construction of the furnace. If the fire brick lining is built perpendicular, the width of grate will be about equal to the diameter of the boiler. On the other hand, if the lining is giving a batter of three inches, starting at the level of the grate, then the width will be reduced six inches. It is customary to allow one square foot of grate surface to every 36 sq. ft. of heating surface. The distance of the

grate-bars from the shell of the boiler varies from 24 to 28 in., according to the dimensions of the boiler.

Blow Off Pipes.—Blow off pipes should always be connected with the lowest part of the water space of a boiler. If there is a mud drum, then of course the blow off should be connected with it; but if there is no mud drum, the blow off should connect with the bottom of the shell, near the back head, extend downwards to the

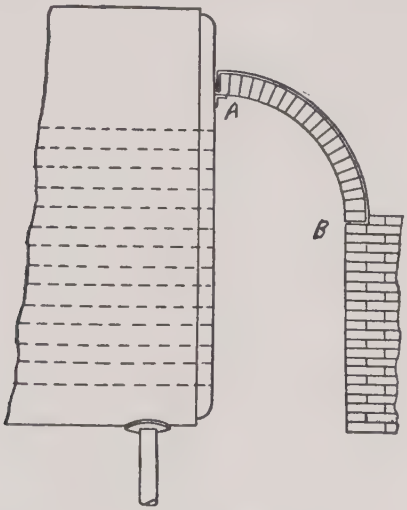


FIG. 124.

floor of the combustion chamber, and thence horizontally out through the back wall, where the blow off cock can be located.

Back Arches.—A good and durable arch can be made for the back connection, extending from the back wall to the boiler head, by taking flat bars of iron $\frac{5}{8}$ x 4 inches, cutting them to the proper length and bending them in the shape of an arch, turning 4 inches of each end back

at right angles, as shown in Fig. 121. The distance O-B should equal that from the rear wall to the boiler head, and the height, O-A, should be about equal to O-B, and should bring the point A about two inches above the top row of tubes. The clamp thus formed is filled with a course of side arch fire brick, Fig. 122, and will form a complete and self-sustaining arch, the bottom, B, resting on the back wall, and the top, A, supported by an angle iron riveted across the boiler head about three inches above the top row of tubes. See Figs. 123 and 124.

Enough of these arches should be made so that when laid side by side they will cover the distance from one side wall to the other across the rear end of the boiler. A fifty-four-inch boiler would thus require six clamps, a sixty-inch boiler seven clamps, and a seventy-two-inch boiler would require eight clamps; the length of a fire brick being about nine inches. In case of needed repairs to the back end of the boiler the sections can be lifted off, thus giving free access to all parts, and when the repairs are completed the arches can be reset with very little trouble and much less expense than the building of a solid arch would necessitate. This form of segmental arch allows ample freedom for expansion of the boiler, in the direction of its length, without leaving an opening when the boiler contracts. It will be seen that within all possible range of boiler movement in either direction the arch will, with this construction, always remain close to the head.

The crosswise construction of arch bars, while affording equal facility in repair work, is necessarily more expensive than the form here described, and is also open to the objection that it cannot follow the contracting boiler and maintain a tight joint or connection between the back arch and the rear head above the tubes.

Suspension Method.—A great improvement is made when we suspend such boilers from I-beams supported by cast iron columns. Figures 125 and 126 show the

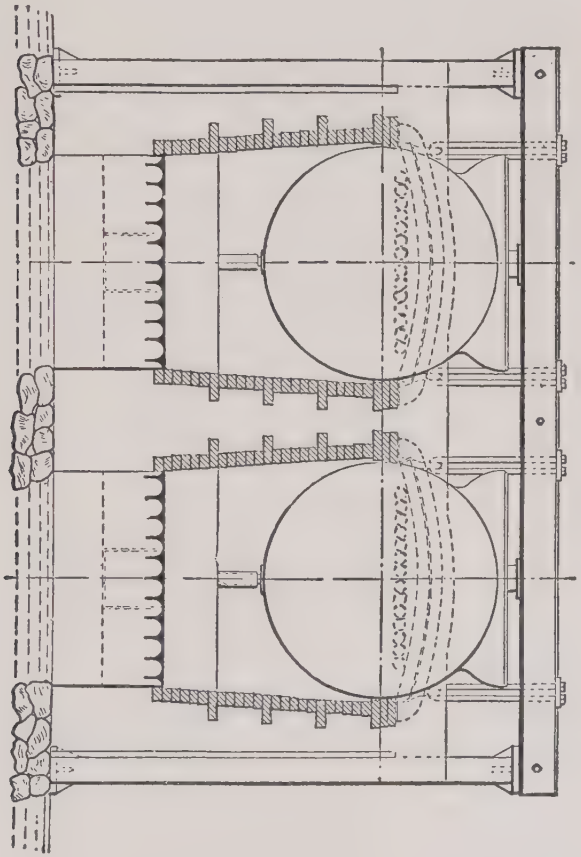


FIG. 125.

setting of this type of boilers either singly or in double batteries, by means of suspension. In setting an even number of boilers, as six or eight in one setting, it is

best to divide them into pairs so that not more than two boilers will be suspended between supports.

The principal reason for this is that when the large sizes, such as from 150 to 250 horsepower, are used, the size I-beam required to safely carry this load between

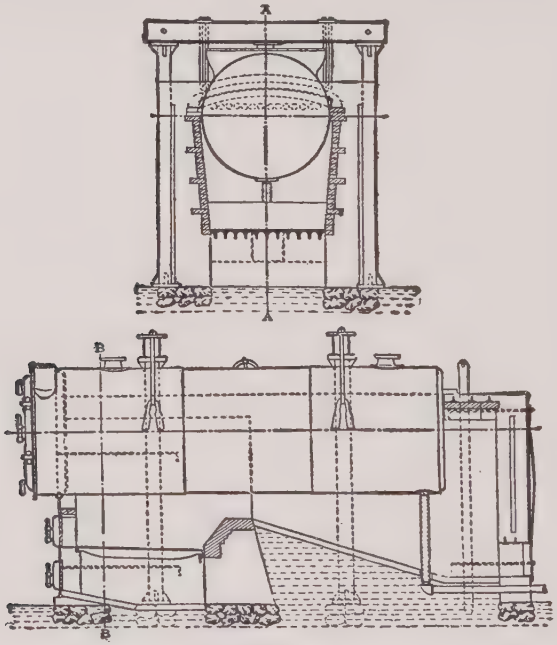


FIG. 126.

supports is so large that it overbalances the cost of two or more cast-iron columns.

In setting an odd number of boilers, such as three or five, in a battery, columns are usually placed between the boilers with a 2-inch air space all around the column, and an air duct at the bottom of the setting which runs through from the front to the back and connects with

each air space around the column. This allows a free circulation of air, thus tending to keep the columns comparatively cool. In setting boilers in this way the columns and I-beams are set in position first. The boiler is then hoisted to the proper height by means of tackle, which is attached to the I-beams, and when the boiler is brought to the proper height, the U-bolts are slipped into place and fastened by nuts and washers to the I-beams.

This method abolishes the use of blocking and leaves all of the space under the boilers clear for the brick work.

The expansion is easily taken care of by the U-bolts and hangers, and if the walls are properly set, they will show no cracks as they carry no weight, and are free to go and come. The accompanying table, No. 35, has been carefully worked out with a factor of safety of 5, and gives the different sizes and lengths of I-beams and columns required for boilers of 36 inches in diameter and 8 feet long, to boilers of 90 inches in diameter and 20 feet in length, giving the total weight to be supported and the sizes, weights, and positions of the columns, and I-beams required:

TABLE 35.

SIZES AND WEIGHTS OF COLUMNS AND I-BEAMS REQUIRED IN SETTING RETURN TUBULAR BOILERS.

HORSE POWER	15	20	25	30	35	40	45	50	60	70	75	80	90	100	125	150	175	200	200	225	225	250
Dia. of boiler in inches	36	36	42	42	44	48	50	54	54	60	60	60	66	66	72	72	78	78	84	84	90	90
Length of tubes in feet	8	10	10	12	12	12	13	13	15	14	15	16	15	16	16	18	18	20	18	20	18	20
Length of curtain sheet in inches	11	11	12	12	12	14	14	14	14	16	16	16	17	17	18	18	18	18	20	20	22	22
Total weight of boiler and water	6500	7500	9400	10500	11500	13300	14200	15300	17800	20800	24800	27200	30300	35000	40000	44000	48000	56000	55000	67000	65000	75000
Rear head to center of hanger	2–0	2–6	2–6	3–0	3–0	3–0	3–3	3–3	3–0	3–6	3–9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
Center to cen. of hang'rs	4–0	5–0	5–0	6–0	6–0	6–0	6–6	6–6	7–6	7–0	7–6	8–0	7–6	8–0	8–0	9–0	9–0	10–0	9–0	10–0	9–0	10–0
Front head to center of hanger	2–0	2–6	2–6	3–0	3–0	3–0	3–3	3–3	3–0	3–6	3–9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
Distance between C of supports (1 boiler)	6–6	6–6	7–0	7–0	7–2	7–6	7–8	8–0	8–0	9–0	9–0	9–0	9–6	9–6	10–0	10–0	10–6	10–6	11–0	11–0	11–6	11–6
Distance between C of supports (2 boilers)	11–8	11–8	12–8	12–8	13–0	13–8	14–0	14–0	14–8	16–2	16–2	16–2	17–2	17–2	18–2	18–2	19–2	19–2	20–2	20–2	21–2	21–2
Length of I-beam for 1 boiler	7–3	7–3	7–10	7–10	8–0	8–4	8–6	8–10	8–10	10–0	10–0	10–0	10–6	10–6	11–0	11–0	11–7	11–7	12–0	12–0	12–6	12–8
Length of I-beam for 2 boilers	12–6	12–6	13–8	13–8	14–0	14–8	15–0	15–10	15–10	17–4	17–4	17–4	18–4	18–4	19–5	19–5	20–8	20–6	21–6	21–6	22–6	22–6
Size of I-beam required for 1 boiler	4	4	5	5	5	6	6	6	6	7	7	7	7	7	8	8	9	9	9	9	9	10
Size of I-beam required for 2 boilers	6	6	8	8	9	9	9	10	10	12	12	12	12	12	15	15	15	15	15	15	15	15
Weight per ft. of I-beam for 1 boiler	7 5	7 5	9 75	9 75	9 75	12 25	12 25	12.25	12.25	15	15	15	15	15	18	18	21	21	21	21	21	25
Weight per ft. of I-beam for 2 boilers	12.25	12.25	18	18	21	21	21	25	25	31 5	31.5	31.5	30	40	42	42	60	60	60	80	80	80
Length of cast-iron col.	8–0	8–0	8–0	8–0	8–8	9–3	9–5	10–0	10–0	10–8	10–8	10–8	11–2	11–2	12–0	12–0	12–6	12–6	13–0	13–0	13–10	13–10
Outside dia. of C. I. col. for 1 boiler	4	4	4	4	4	5	5	5	5	5	5	5	6	6	6	6	6	6	6	6	6	6
Outside dia. of C. I. col. for 2 boilers	5	5	5	5	5	6	6	6	6	6	6	6	6	6	8	8	8	8	8	8	8	8
Size of flange on ends of col. for 1 boiler	9½	9½	10	10	10	10½	10½	10½	10½	11½	11½	11½	11½	11½	12	12	12½	12½	12½	12½	12½	13½
Size of flange on ends of col. for 2 boilers	10½	10½	12	12	12½	12½	12½	13½	13½	14	14	14	14½	14½	15	15	16	16	16	17	17	17
Thickness of C. I. col. for 1 boiler	½	½	½	½	½	½	½	½	½	⅝	⅝	⅝	⅝	⅝	⅝	⅝	1	1	1	1	1	1
Thickness of C. I. col for 2 boilers	½	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝	1	1	⅝	⅝	⅝	⅝	1	1	1	

Insulation.—All boilers should be well protected from the cooling influence of outside air, if economy of fuel is any object. The tops of horizontal boilers should be covered with some kind of heat insulating material, or arched over with common brick, leaving a space of two inches between the boiler and the arch. The resulting saving in fuel will far more than compensate for the extra expense in a very short time. All cracks in the side and rear walls should be carefully pointed up with mortar or fire clay. One source of heat loss in return flue boilers is short circuiting from the furnace to the breeching, caused by the arches over the fire doors becoming loose and shaky, and allowing considerable of the heat to escape directly to the stack instead of passing under the boiler and through the tubes. Another bad air leak often occurs at the back connection when the arch rests wholly upon iron bars imbedded in the side walls. This leak, as has already been noted, is caused by the expansion of the boiler, which gradually pushes the arch away from the back head until, in the course of time, there will be a space of $\frac{5}{8}$ inch and sometimes $\frac{3}{4}$ inch between the head and the arch. The obvious remedy for this is an arch that will go and come with the movement of the boiler, and such an arch can be secured by building it in sections, as illustrated by Figs. 122 and 124.

Coal and Ash Handling.—The question of handling fuel and ashes should always be considered when locating the steam plant. No matter how well placed the plant may be, there should be provided adequate means for handling coal and for removing the ashes. Various conditions require vastly different methods, but there are very few plants where the conveyor-elevator cannot be profitably installed for bringing in coal and carrying out ashes, the same apparatus serving both opera-

tions, and permitting the coal storage and ash dump to be located at will, either on the ground adjacent to, or removed from the building, or overhead in, or outside the boiler house.

The first cost of an installation of this kind is comparatively low, and once installed it will run for years with little cost for repairs or renewal. Therefore, apparatus of this kind should always be considered in

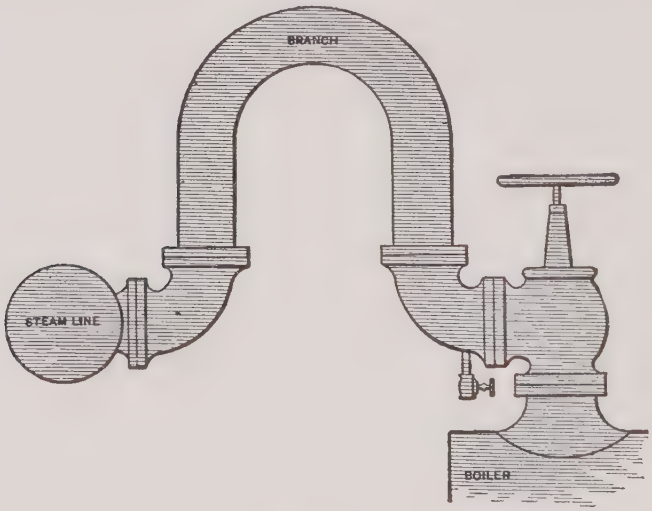


FIG. 127.

connection with the installation of a battery of boilers.

Smoke Stacks.—The smallest opening of the chimney should be one-fourth of the grate surface, and its height about 25 times this smallest opening, but never under 50 feet.

Chimneys which are built without means for ascending on the outside must have an opening at the top of not less than 24 inches. The lower diameter of a brick chim-

ney should be 0.02 of the height of the chimney larger than that of the top opening.

Round chimneys are best because they offer less resistance to the wind, do not retard the revolving action of the gases, and having the smallest circumference for a given diameter, suffer the least from loss of heat. The bottom of the opening of the chimney should be at least 24 inches to 30 inches from the bottom of the bridging

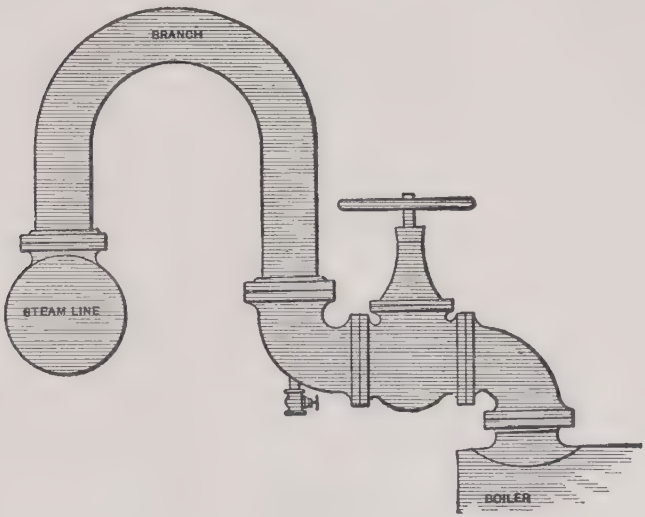


FIG. 128.

and if more than one bridging enters the chimney they should be so arranged as to give the gases from all of them the same direction, as otherwise a considerable loss of velocity would occur.

Brick chimneys are best, but also the most expensive. They last long and need no painting, nor much repairs, except pointing up once in a long while. They retain the heat much better than iron smoke-stacks.

Iron smoke-stacks are made either self-sustaining, or are held by guy ropes. In the first case, the stack is provided with a strong cast-iron base, bolted down upon a foundation heavy enough to prevent the strongest wind overturning the stack. The stack is also made of heavier iron. The necessity of keeping in paint, and the danger of rusting on the inside if not continually in use, are two disagreeable conditions connected with the use of iron chimneys.

In many places iron stacks are preferred to brick chimneys. Iron stacks require to be kept well painted to prevent rust, and generally, where not bolted down, they need to be braced by rods or wires to surrounding objects. With four such braces attached to an angle iron ring at 2/3 the height of stack, and spreading laterally at least an equal distance, each brace should have an area in square inches equal to 1-1000 the exposed area of stack (diam. $\times$ height) in feet.

Stability, or power to withstand the overturning force of the highest winds, requires a proportionate relation between the weight, height, breadth of base, and exposed area of the chimney. This relation is expressed in the equation

$$C\frac{d\,h^2}{b} = W,$$

in which d = the average breadth of the shaft; h = its height; b = the breadth of base,—all in feet; W = weight of chimney in lbs., and C = a co-efficient of wind pressure per square foot of area. This varies with the cross-section of the chimney, and = 56 for a square, 35 for an octagon, and 28 for a round chimney. Thus a square chimney of average breadth of 8 feet, 10 feet

wide at base and 100 feet high, would require to weigh $56 \times 8 \times 100 \times 10 = 448{,}000$ lbs. to withstand any gale likely to be experienced. Brickwork weighs from 100 to 130 lbs. per cubic foot, hence such a chimney must average 13 inches thick to be safe. A round stack could weigh half as much, or have less base.

When two or more boilers are set in battery, there should be a separate damper to each boiler and another damper in the stack or main smoke flue, and this dam-

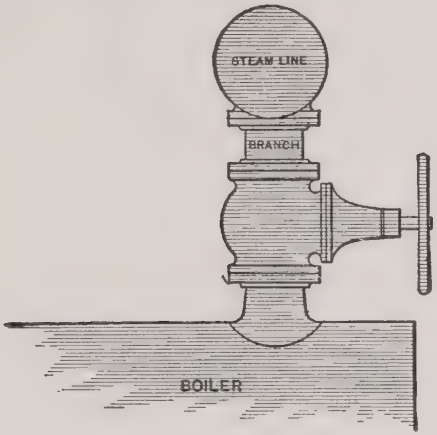


FIG. 129.

per should be connected to a good damper regulator which is actuated by steam or water pressure, and which is controlled by the steam pressure in the boilers.

Connecting Boilers to Main Header.—The design of the branch pipe from each boiler to the main steam header is one of the most important details pertaining to boiler setting. The first matter to be determined is the number and kind of valves to be used. The most common practice is to use one globe valve, but the only argument in favor of this is the lower first cost. It is

better practice to use two valves in this branch for considerations of safety and convenience. Surely no one should be placed in jeopardy of such a horrible death as scalding in a boiler, for the sake of the first cost of an extra stop-valve. Then boilers have been known to go for months in need of inspection and repair in the drums, simply for the reason that it was impossible to do the

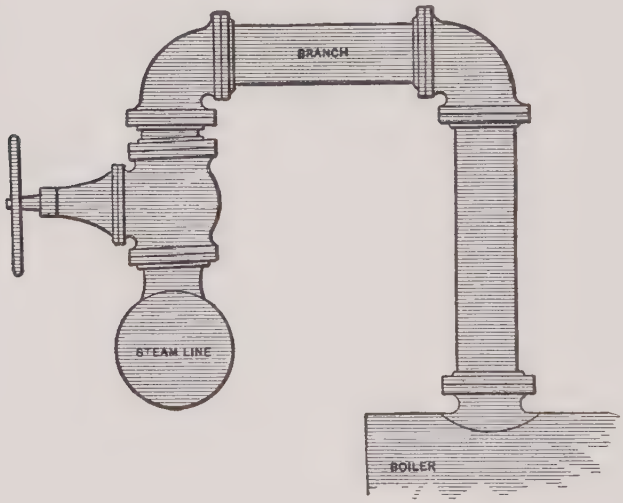


FIG. 130.

work, owing to the fact that the crown valve was leaking, and no one could safely enter the drums. The single valve with its usual leaks is not a safe proposition. Therefore, there should be two valves located in each branch.

Regarding the kind of valves to be used, it appears to be universally conceded that gate valves are the most desirable for this purpose, owing to their strength and

durability. A gate valve will also open more gradually, which is a valuable feature in turning on the steam.

Another reason for preferring the gate-valve is that there is no possibility of a careless or ignorant workman putting it in so that, when shut, the pressure is against the stem tending to strip the threads, as has been done with globe valves.

The objection to the use of angle-valves, however, is

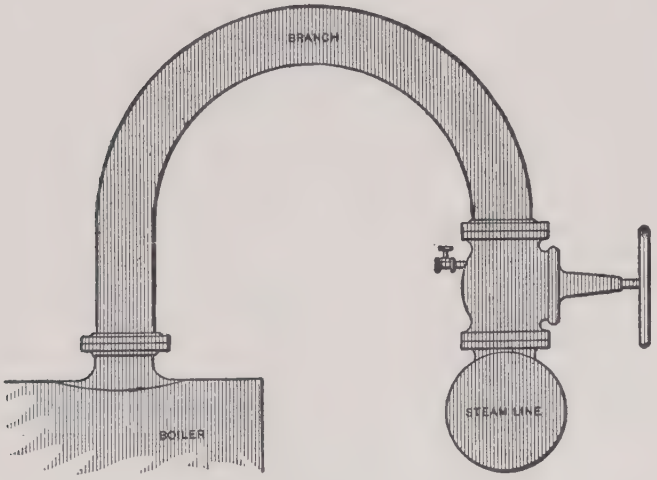


FIG. 131.

much stronger than to globe valves, first because they have the same rapidity of opening as the globe valve, and, second, especially because the direction of flow must be changed in the valve and it is therefore a form of construction peculiarly liable to injury by water ram, if there be any chance for water to collect in the pipe.

Some of the particular points to be observed in the arrangement of the branches in order to provide as far as possible for safety are: first, to avoid the use of

angle valves; second, to arrange the piping in such a manner as to prevent pockets in which condensed steam may gather; third, to make provision for the free expansion and contraction of the pipes.

In any arrangement of piping in which water is liable to pocket, proper drip pipes should be placed where necessary, and the water kept drained out. Figs. 127 to 131 show connections which will give good service. Fig. 131 could be improved by placing another gate valve between the bend and the boiler.

UNLOADING AND SETTING UP MACHINERY.

The work of unloading machinery from car or boat and getting the machines into the building, and onto their foundations is generally performed by a gang of riggers, the millwright himself, or someone of his assistants superintending the job. In fact a millwright, in order to be successful in his chosen occupation, should be well posted in the details of handling machinery to the best advantage.

The most practical method of unloading and moving heavy machines or parts of machinery, is by means of the skid and roller, enough of these being placed underneath the machine to hold it steady while being moved to its proper location. The machines are lifted, and placed upon the skids by means of the bar, lever and bait; the screw jack also playing an important part in this work. The hydraulic jack, or "whisky-jack" as it is called, is also a very useful appliance in millwright work. A few words of explanation regarding its action may not be amiss here.

The hydraulic jack is a small hydrostatic press made in portable form, the pump being contained inside the

plunger, the cylinder forming the body of the jack. Alcohol is used as the liquid for operating the hydrostatic part of the jack. Water may be used instead of alcohol, but it is liable to rust some of the parts, and in cold weather it would freeze, thus causing trouble. To operate one of these jacks, insert the lever so that the shoulder on one side of the lever will project downward and strike a lug on the head of the jack. An ordinary pumping movement of the lever will then force the liquid out of the piston into the reservoir underneath it, causing the jack piston and head to rise according to the bulk of liquid displaced. To lower, reverse the lever, turning the lug uppermost, then bear down on the lever and the valve in the pump will be forced to open, allowing the liquid to pass back into the piston reservoir. In some jacks this form of construction is reversed, the piston standing upon the foot or base and the cylinder being attached to the head of the jack. Sometimes the arrangement of the pump lever is opposite that described above, the lug being placed uppermost, and a lifting motion being necessary to lower the jack after the lever has been reversed.

The Ball-Bearing Screw-Jack.—There is also a most excellent form of jack known as the "ball-bearing screw-jack," in which there is a set of balls between the end of the lifting screw and the head of the jack. As friction usually consumes more than one-half the power applied to the plain screw-jack, the value of the ball arrangement is obvious. In this jack there is a pair of miter gears with a ratchet arrangement on the power lever, thus making it possible to "pump" this excellent form of jack the same as the whisky-jack, and the ratchet attachment permits the jack to be worked in restricted quarters equally as well as the other.

In operating any and every form of jack, the one point to be closely watched is the distance which the jack has been extended. In the whisky-jack, too great an extension will permit the cylinder to be scored by the piston or plunger, and the jack becomes useless until machine-shop repairs can be made. In any form of screw-jack, the same danger exists, but if the jack be extended too far, the thread may be stripped off the screw or out of the nut.

When jacking up machinery every precaution should be observed, and the machine should be followed up closely with substantial blocking in order to guard against any possibility of the accidental dropping of the machine as much damage might be done, and possibly, loss of life occur. When a machine has been safely placed upon skids and rollers, it should not by any means be moved until the pull tackle, and the back haul have both been attached. The pull may be either a differential chain tackle, a heavy rope tackle, or if the machine to be moved is very heavy, a cable and winding drum may be used. For the back haul a heavy rope tackle is preferable, although, in the absence of the tackle, a plain hitch may be made of a heavy rope which is then snubbed around a post or any convenient object that is permanent.

The rigger gang should be under the immediate direction of a competent "boss" or foreman, whose word of command must be the law. The majority of laborers are naturally careless, and are apt to take needless risks. Therefore discretion should be observed in the selection of the rigger gang, and especially in choosing the foreman thereof.

Erecting Machines Upon Foundations.—The building of foundations for machines has already been discussed

and will not be enlarged upon here, except to say that when making concrete foundations for machinery, use a bank or pit sand whose grains are angular or sharp. If mixed with clay or loam the sand must be washed before using. The gravel used in this work is usually found with the sand. In good gravel the stones are hard, and

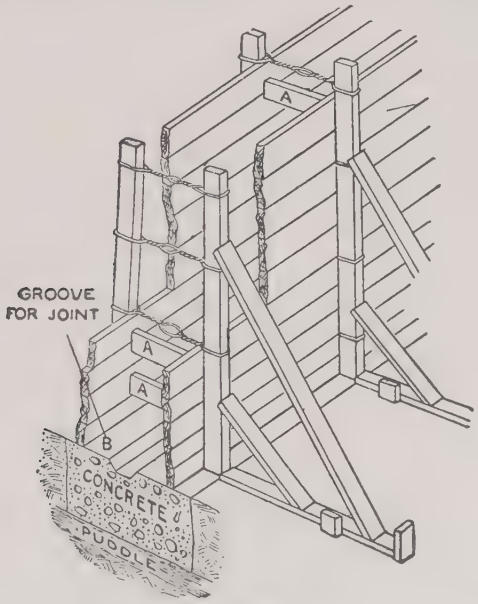


FIG. 132.
FORM OF CONCRETE BASE.

irregular in shape and size. Broken stone makes better concrete than gravel, but is more expensive. The stone should be graded from ¼ to 1 inch in diameter. The cement used should be of good quality. Mix thoroughly; better have too much water than too little.

If possible, foundations should rest on hard pan, gravel, or hard clay. Made ground should be shoveled

out for a depth of at least 12 inches below the bottom of the foundation and rammed as hard as possible. The 12 inches should then be filled with clay puddle, well rammed in place. Of course the forms must be strong, tight and smooth on the inside. A good form is shown in the cut, Fig. 132. It will be noted that the wall is braced in three places, besides being wired. The wire is about No. 12 iron wire. The braces A are loose, or only tacked in place, and are removed as the wall is built up. Do not nail the sheeting too securely to the studding. Any carpenter can put these forms up and the cheap, unskilled men that wheel the concrete may be employed to assist in the bracing and sheeting. Different methods of bracing the forms will readily suggest themselves to the mechanic in charge, to conform to the local conditions. The cut shows the wall with a footing extending several inches outside it.

Care should be exercised in placing machines upon ready-built foundations to prevent the cracking or crumbling of any portion of the concrete or masonry. The masonry or concrete should be given ample time to set and become solid. Machines generally require to be leveled up after landing them on their foundations, and to do this it becomes necessary to raise one side, or one end, or perhaps just one corner of the base plate a small fraction of an inch, and this usually must be done without the aid of a jack, for the reason that there is no chance for getting the jack underneath the machine, or any part of it that will withstand the pressure.

In such cases a four-sheave tackle should be rigged over head on a beam or girder. A differential chain block will often answer the purpose for light machines. By the use of the four-sheave tackle, and a winch no

strain whatever need be brought to bear upon the foundation by means of jacks and levers.

Lever and Cable-Hoist Lift.—When the heavy end of a machine must be raised as above described, and it is feared that the weight is beyond the capacity of any hoist available, then resource may be had to the combined lever and cable hoist shown by Fig. 133. This device may be used equally well with a jack or the differential chain hoist as with the wire-cable business. As

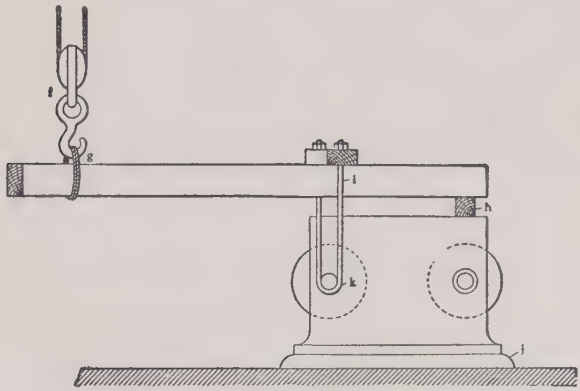


FIG. 133.
LEVER, AND CABLE HOIST.

shown, it consists of a heavy timber, one end of which is attached by a sling g to the lower block of the cable tackle. The other end of the timber rests upon a block placed across the top of the machine, as at h. At the other end of the machine—the end to be raised—the timber is made fast by means of a chain or a yoke to some convenient shaft or projecting portion of the frame, as at k. A pull at f will result in raising a load at k heavier than 4,000 pounds in the ratio of the horizontal distances g k and k j. Thus, if k g be twice k j, then our limit

force of 4,000 pounds at f will handle 8,000 pounds at k, thereby enabling a pretty heavy load to be lifted by 32 pounds exerted on the cranks of the wire-cable hoisting device.

Leveling and Setting the Machine.—After elevating the end or side as it may be, a sufficient distance to bring the machine level, steel wedges should be placed at intervals under the base of the machine, and the space between machine and foundation be filled with cement, brimstone, or plaster of paris, or calcined plaster, which latter makes a very good filling if properly mixed and allowed a sufficient time for setting (24 hours or longer if possible), before the weight is allowed to rest upon it.

Plaster to be poured should be mixed with more than one-half its weight of water. That amount gives a paste which can be shaken from a trowel but which will not drop without shaking. Ascertain by trial how much more water is needed to make up plaster which will flow freely, then weigh out charges of plaster and of water and have them at hand ready for use. After the clay or putty dams have been placed around the machine footing, mix together the water and the plaster, each charge ranging from one pound to ten pounds according to the size of the job. It is better not to mix more than ten pounds of plaster at a time. Mix the water and plaster as quickly as possible, and pour at once.

If brimstone is used for filling, it should be melted in a vessel having a tight fitting cover. The fire should be light, as brimstone ignites easily, and if allowed to remain over the fire too long after being once melted, it is apt to become waxy, and will not flow readily.

If cement, or plaster of paris is used under a machine, a piece of wire should be used for the purpose of

pushing or carrying the running cement into all of the remote corners and cavities in order that they may be filled with cement or plaster, thus securing a firm solid foundation.

Knots and Hitches.—While on the subject of handling,

FIG. 134.

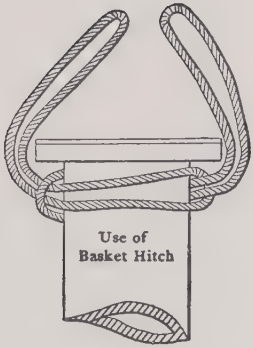


FIG. 135.

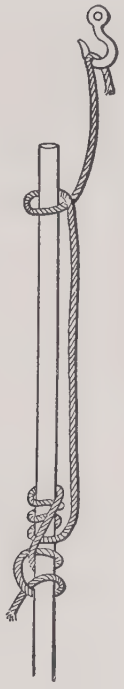


FIG. 136.

and placing machinery, a short space will be devoted to the discussion and illustration of the various kinds of knots and hitches in use by riggers.

Fig. 134 made of two endless slings, and used as shown in Fig. 135, is a reliable and useful basket hitch provided the slings are of equal length, or with one sling

just long enough to encircle one-half the circumference of the cylinder, and the other long enough to pass through both loops of the short sling, and have its own loops catch the tackle. A shaft may be hoisted endwise with this hitch by using a collar, or lathe dog for a safety stay, but a better hitch for hoisting a shaft endwise is the biting-rolling hitch shown in Fig. 136. Sometimes when a ready made endless sling is not available, a flat knot is made in a short length of rope and used

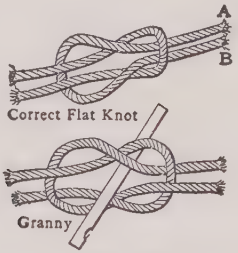


FIG. 137.

FIG. 138.

in place of the sling. The correct form of flat knot is shown at the top of Fig. 137, while the incorrect form, or "granny" knot, is shown below. This latter is unsafe, and should be avoided at all times. Fig. 138 shows a good and safe method of shortening a long rope. This is known as a sheep-shank, and it is evident that any amount, and any length of loop may be used, but it should be remembered that the over-lap at x, x, should be at least 6 inches in length to insure absolute safety.

The hitch illustrated in Figs. 139 to 142 is made without passing the end of the rope, and provides two loops to which a tackle block may be hooked. Fig. 139 shows the start; Fig. 140, the second stage; Fig. 141, the manner of rolling the two loops into the standing portion of the rope, and Fig. 142 shows the two loops x x brought vertically down (after rolling) and ready for service. The block or fall must be hooked into both loops. This is a safe and reliable hitch that can be wiggled in at

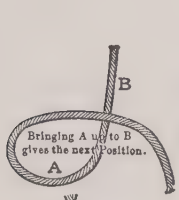


FIG. 139.

FIG. 140.

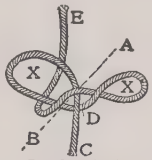


FIG. 141.

FIG. 142.

FIG. 139.

FIG. 143.

FIG. 144.

FIG. 145.

FIG. 146.

any point in a rope, and besides being perfectly reliable, it is easily, and quickly made and unmade.

Fig. 143 is an old and well known friend of the rigger. Bow lines are shown in Figs. 144 to 146. In making these knots the slack line x may go either in front of, or back of the standing rope y, as shown in Figs. 144 and 145, but in either case it must be passed through the loop z after going around y. This is shown in Fig. 146, which is an illustration of a completed bow line. Fig. 147 shows how a split hammer handle, or split monkey

wrench handle may be temporarily repaired by the endless wound splice. The method of making it up is clearly shown, and it is evident that by pulling at A, the loop B will make a similar loop in D at C, and continued pull will draw the crossed loops out of sight. The loop ends may then be cut off closely. The knot shown completed in Fig. 148 is known as a jury mast knot, also as a "bottle hitch." It can be used in place

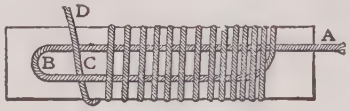


FIG. 147.

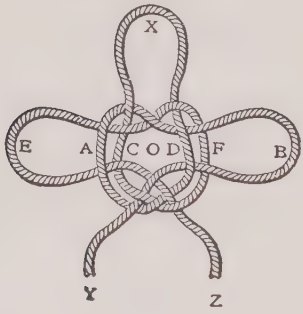


FIG. 148.

of a mast iron, at the top of a derrick, or gin pole to make guys fast to.

It appears at the first glance to be very complicated, but on the contrary it is very easily made. To practice making it, take a piece of strong cord between the thumb and forefinger of each hand, with a space of about 6 inches between the hands. Then twist the cord right-handed with the thumb and forefinger of the right hand only. This will throw up a "bight" like Fig. 149 with the part A under B. Next grasp the loop between

the thumb and forefinger of the left hand at the point where the two parts cross. Then move the thumb and forefinger of the right hand along the cord about 6 inches, and throw up another "bight," laying it on top of the first one. This will form Fig. 150. Hold these two "bights" with the left thumb and forefinger, measure off another 6 inches with the right hand, and throw the last "bight." Place it on top of the last one made. This will form a combination, such as shown in

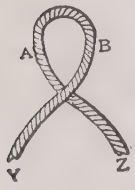


FIG. 149.

FIG. 150.

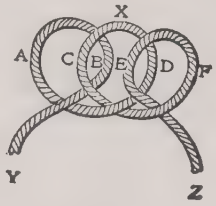


FIG. 151.

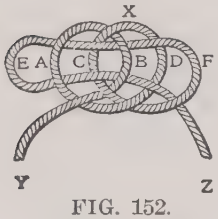


FIG. 152.

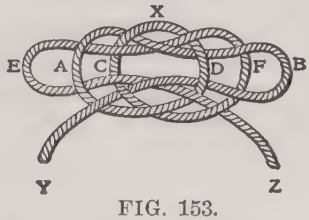


FIG. 153.

Fig. 151. Then take the part E in the last "bight" in Fig. 151 and, while holding the other parts in place, pass it under B, over C, and under A. .

This makes Fig. 152. Then take B, Fig. 152, and pass it under D, and over F, the result being Fig 153. Now while holding E in the left hand, and B in the right, have a helper take hold of x and pull it. This will form the completed knot as shown in Fig. 148. In practice, the central loop O is placed over the reduced

top of the mast, or gin pole. The forestay or guy, is made fast to x, the front stays to E, and B, while y, and z form the back stays. Any strain on the stays serves to draw up and tighten O. By pulling on y and z in opposite directions, the knot is easily loosened, and comes out.

Fig. 154 shows how to sling a barrel for hoisting. With both heads of the barrel on, and the bung in place, this is a comparatively easy matter, but with one head out, it is not so easy.

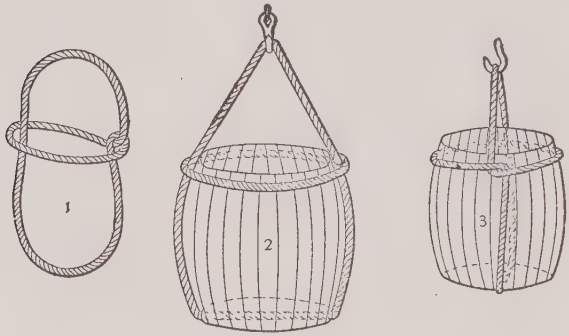


FIG. 154.
SLINGING A BARREL.

The illustrations 1, 2 and 3, in Fig. 154, show how it may be accomplished in a very neat, and easy manner.

Splicing Rope.—The splicing of a rope, especially when used for the transmission of power, is an important matter. The principal points upon which the success of the splice depends are: first, the length of the splice, which depends of course upon the diameter of the rope, and is given in table 36; second, the diameter of the splice, which should be the same as the diameter of the rope; third, the proper securing of the ends

of the strands of the splice, which must be fastened in such a manner that they will not wear, or whip out, or cause the overlying strands to unduly wear; fourth, the workmanship of the splice, which should be the best that it is possible to secure.

When splicing an old and a new piece of rope, the new piece should be thoroughly stretched, for, at best, it is an exceedingly difficult task on account of the stretch and difference in diameter of the rope.

The following instructions, and illustrations for

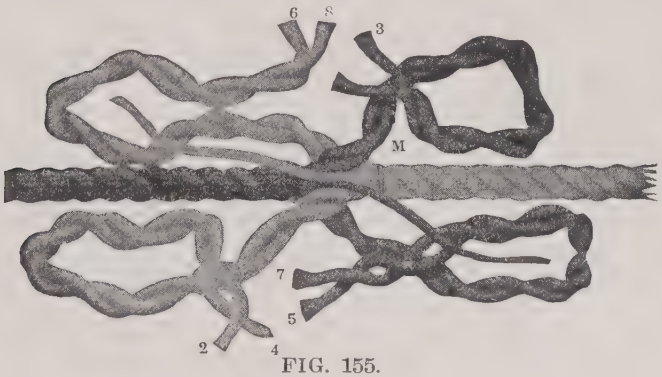


FIG. 155.

making standard rope splices are supplied by the American Manufacturing Company, in their "Blue Book of Rope Transmission." There are many different splices now in use, but the one that experience has proved best is what is known as the English transmission splice.

In describing this we take for our example a four-strand rope 1¾ inches in diameter, as spliced on sheaves in the multiple system. The rope is first placed around the sheaves, and, with a tackle is stretched and hauled taut. The ends should pass each other from six to seven feet, the passing point being marked with twine on each

rope. The rope is then slipped from the sheaves, and allowed to rest on the shafts, to give sufficient slack for making the splice. Unlay the strands in pairs as far back as the twines M, M, Fig. 155, crotch the four pairs of strands thus opened, cores having been drawn out together on the upper side.

Then having removed marking twine M, unlay the two strands 6 and 8 (still in pairs), back a distance of two feet, to A; the strands 1, and 3 (also in pairs), being carefully laid in their place. Next unlay strands 5 and 7 in pairs, to A′, replacing them as before, with strands

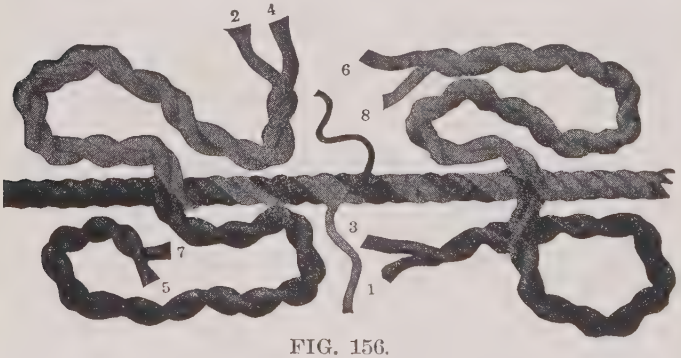


FIG. 156.

2 and 4. The rope is now as shown in Fig. 156. The pair of strands, 6 and 8, are now separated, and 8 is unlaid four feet back to B, a distance of six feet from center, strand 6 being left at A. The pair of strands 1 and 3 having been separated, 3 is left at A, as companion for 6, strand 1 being carefully laid in place of strand 8 until they meet at point B. The two pairs of strands 2-4 and 5-7, are now separated and laid in the same manner, every care being taken, while thus putting the rope together, that the original twist and lay of strand is maintained. The protruding cores are now

cut off so that the ends, when pushed back in rope, butt together. The rope now appears as shown in Fig. 157, and after the eight strands have been cut to convenient

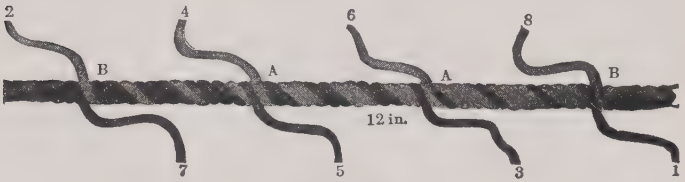


FIG. 157.

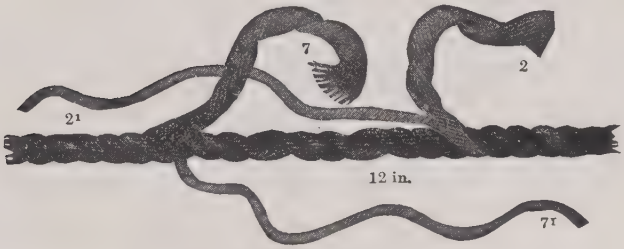


FIG. 158.

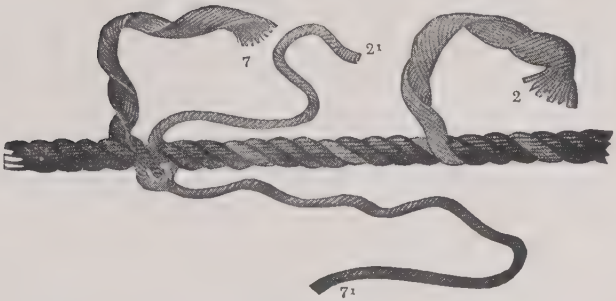


FIG. 159.

working lengths (about two feet), the companion strands are ready to be fastened together and "tucked." This operation is described for strands 2 and 7, the method being identical for the other three pairs.

Unlay 2 and 7 for about twelve to fourteen inches, divide each strand in half by removing its cover yarns, see Fig. 158, whip with twine the ends of interior yarns 2′ and 7′; then leaving cover 2, relay 2′ until near 7 and 7′, here join with simple knot, 2′ and 7′, Fig. 159. Divide cover yarns 7, and pass 2′ through them, continuing on through the rope under the two adjacent

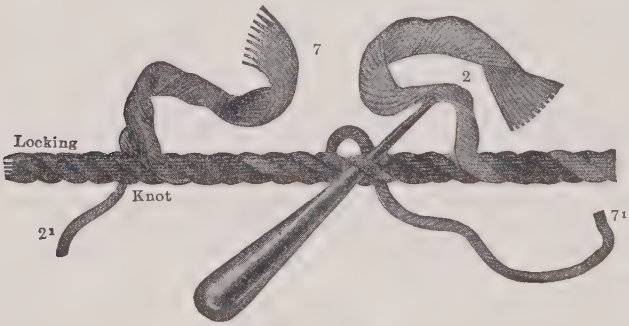


FIG. 160.

FIG. 161.

strands, avoiding the core, thus locking 2′, Fig. 160. In no event pass 2′ over these or any other strands.

Half strand 7′ must now be taken care of. At the right of the knot made with 2′ and 7′, 2′ is slightly raised with a marlin spike, and 7′ passed or tucked around it two or three times, these two half strands forming in this way a whole strand.

Half strand 7′ is tucked until cover 2 is reached, whose yarns are divided, and 7′ is passed through them, and

drawn under the two adjacent strands, forming again the lock.

The strand ends at both locks are now cut off, leaving about two inches, so that the yarns may draw slightly without unlocking. This completes the joining of one pair of strands. See Fig. 161. The three remaining pairs of strands are joined in the same manner.

After the rope has been in service a few days, the projecting ends at locks wear away, and if tucks have been carefully made, and the original twist of yarns preserved, the diameter of the rope will not be increased, nor can the splice be located when the rope is in motion.

As before stated, Table 36 which follows, contains useful data regarding the splicing of manilla transmission rope, also approximate weight in pounds per lineal foot, of rope of various diameters, together with the breaking strength, maximum allowable tension in pounds, and smallest diameter of sheaves over which the different sizes of rope may be run with safety.

TABLE 36.—MANILLA TRANSMISSION ROPE. SPLICING AND SHEAVES.

Diameter of Rope in Inches	Square of Diameter	Approximate Weight per Foot, Pounds	Breaking Strength, Pounds	Maximum Allowable Tension, Pounds	Length of Splice in Feet			Smallest Diameter of Sheaves in Inches	Maximum Number of Revolutions per Minute
					3-Strand	4-Strand	6-Strand		
[illegible]	.25	.12	1750	50	6			20	1060
[illegible]	.2906	.16	2730	80	6			24	970
[illegible]	.5625	.20	3950	112	6	8		27	760
[illegible]	.7656	.26	5400	153	6	8		32	650
1	1.	.34	7000	200	7	10	14	36	570
1¼	1.2656	.43	8900	253	7	10	13	40	510
1¼	1.5625	.63	10,900	312	7	10	16	45	460
1½	2.25	.77	15,700	450	8	12	18	54	380
1¾	3.0625	1.04	21,400	612	8	12	18	63	330
2	4.	1.36	28,000	800	9	14	20	72	290
2¼	5.0625	1.73	35,400	1012	9	14	20	81	255
2½	6.25	2.13	43,700	1250	10	16	22	90	230

SHIFTING BELTS AND BELT SHIFTERS.

In all shops and mills the nature of the work requires more or less stopping, and starting of the various machines, and in many cases the reversal of the same. This necessitates shifting belts, and loose-pulleys, together with belt shifters, the reversal of the machines being accomplished by crossed belts.

Shifting belts are not so much in vogue now as they formerly were, they having been to a great extent displaced by the rim-friction clutch, a very useful and

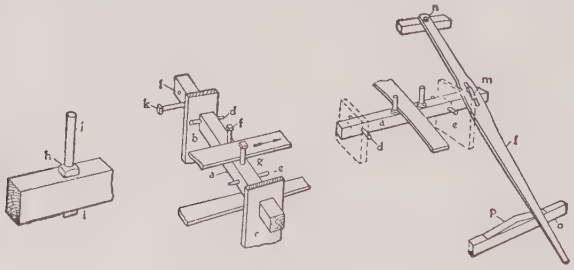


FIG. 162.
ORDINARY BELT SHIFTER.

convenient device. Nevertheless shifting belts are still much in use, and as long as they continue to be used, just so long will belt shifters be required, therefore a few words on this subject will no doubt be appreciated. In the first place a belt shifter should not touch the belt except during the actual time occupied in shifting the belt from one pulley to the other. This means that whether the belt is running on the tight pulley, or on the loose pulley, the shifter should not touch the belt, nor be made, as too many of them are, to do duty as a belt guide to keep the belt on the pulley. When a belt requires a guide to keep it running fair on the pulleys, it

is a pretty fair indication that the shafting is out of alignment. Fig. 162 shows the construction of an ordinary belt shifter. The piece of 2 x 4 inch scantling a is supported in the guides b and c, through which it is free to move back and forth.

The pins d and e, prevent too much movement of the shifter bar a, while the lag screws f and g, serve to force the belt from one pulley to the other when the bar is moved endwise.

In the illustration the lag screws f and g, are shown as quite close to the belt which is wrong. There should be 1 inch or 1¼ inches more width between these screws, or guide pins than the total width of the belt for all belts of four inches or less in width, and for all belts wider than four inches, there should be an increased amount of clearance between belt and guide pins. A better arrangement than lag screws f and g, is shown at h and i, in which the screws are replaced by ¾-in. rods or even larger in diameter to prevent cutting the edges of the belt. At k is shown a pin passing through a hole in the shifter bar close up to the guide board b.

The apparent purpose of this pin is in connection with pin d on the opposite side of board b, to firmly hold bar a in place after the belt has been shifted, thus making it in conjunction with pins f and g, act as a belt guide, which is contrary to good practice, besides making it necessary for a man to go to the shifter bar each time the belt is to be shifted from pulley to pulley. A much better device is a shifter handle shown at l, Fig. 162, made of sufficient length to reach to the point most convenient for operating it. This handle may be made of 2 x 6 inch spruce or pine lumber, having the lower end tapered to about 2 inches in diameter, while it is

left full width at m, and tapered again to about 4 inches at n, where it is bolted to some solid point of support A slot is made in the handle at m, and a bolt or lag screw is used at that point for connecting it to the shifter bar. The method of correctly attaching the two parts is as follows:

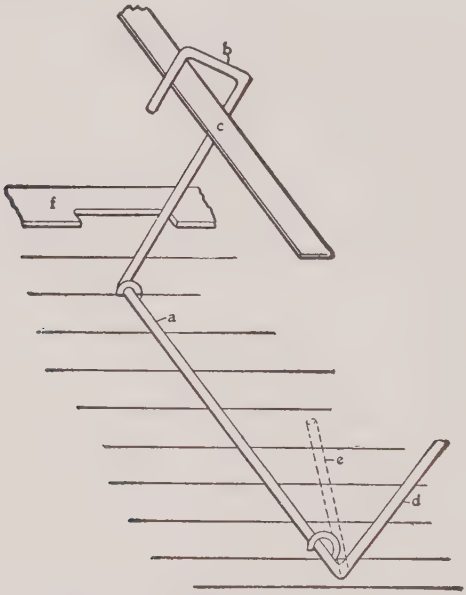


FIG. 163.
ROD BELT SHIFTER.

Place the shifter bar midway of its travel, the belt being half way on each pulley. Then place the shifter handle at right angles to the shifter bar, and while in that position mark the hole for m on the shifter handle. This method provides the same angularity for the handle at each end of its throw. A support o having a double wedge p is placed in a position midway of the

throw of the lever and there fastened permanently thus forming a support for the handle.

Another style of belt shifter called the rope and rod shifter, is shown in Fig. 163. This device is very easily made, as will be apparent upon an inspection of Fig. 163. It consists of a rod of 7/8 or 1 inch round iron, bent to the shape shown at a, b, and d in the cut. The

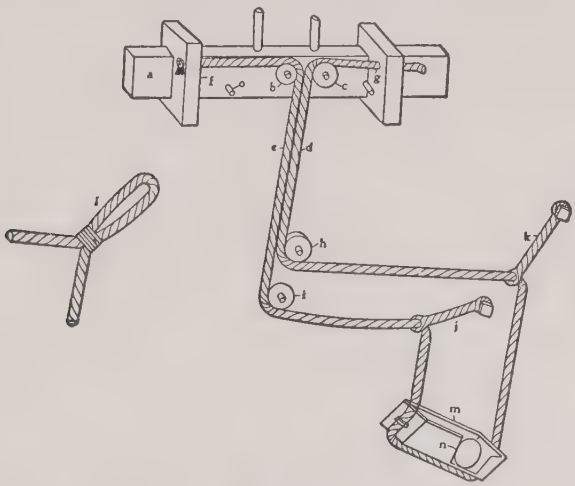


FIG. 164.
ROPE CONTROLLED BELT SHIFTER.

belt is shown at c. The part d of the rod serves the purpose of a handle for operating the shifter.

At f is a double stop intended to prevent the shifter from moving too far in either direction. The combined weights of the yoke b, and handle d, serve to hold the shifter in whichever side of the vertical center of its throw it may chance to be.

It is said to be a very efficient shifter, and may be constructed so as to be operated by ropes connected to

an eye formed in the end of handle d, and leading in opposite directions, and passing over sheaves, thus making it possible to manipulate the shifter from a remote distance. By applying a weighted tension pulley at the farthest point reached by the ropes, the problem of distant belt control becomes easy. Fig. 164 shows one of several methods of arranging a rope control for a belt shifter. The shifter bar a, is fitted with a pair of sheaves 6 inches in diameter, and placed just far enough apart to allow the ropes to pass each other, thus preventing their running off the sheaves. The ends of the ropes are passed through the guides, and knotted thus securing them at that end. By a downward pull on rope e, the shifter bar is forced to the left, and by pulling down on rope d, the opposite effect is produced and the shifter bar is moved to the right, the two sheaves moving with the bar, while the pins o and g in the bar prevent it from moving too far in either direction, by engaging with the guides. The ropes are carried to their desired location over other sheaves placed as shown at h and i. As many of these sheaves may be used as is found necessary. The ropes at the operating end should hang in a vertical position, and a sufficient tension of some sort should be applied at these ends to hold all parts taut, and in their places.

A very convenient appliance for this purpose may be made by attaching a box to the ends of the ropes by means of clamps as shown at m, but in such a manner that the box will stand in a horizontal position when the shifter bar is at mid-travel. A heavy iron ball is placed inside the box as shown at n, where the ball is seen at one end of the inclined box, while at the same time the shifter bar is at the end of its travel in that direction. In order to throw the shifter bar in the other

direction, all that is necessary is to raise the lower end of the box which will cause the ball to roll to the other end of the box, which will then exert the pull on the other rope, and the shifter bar will be moved to the opposite position and held there.

If it is not convenient to obtain a ball for the purpose, dry sand may be used in the box, although it is much slower than a ball, and requires a larger box.

Removing Pulleys from Shafting.—When a solid pulley is to be removed from a section of shaft, sledge hammers should not be used on the hub or arms, as cast iron pulleys are very liable to break or crack under repeated blows.

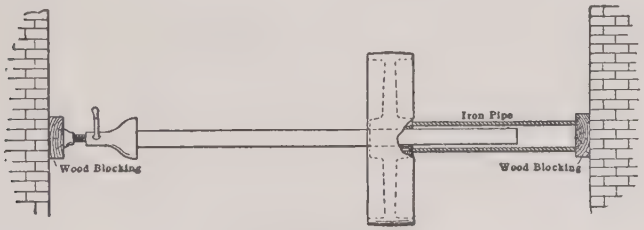


FIG. 165.

One method of removing a pulley is illustrated in Fig. 165, and may be applied when a location can be found between two solid walls that will admit the necessary apparatus. As will be observed, the method consists in forcing the shaft through the pulley by means of a screw jack pressing against the end of the shaft, the pulley itself being held stationary by a piece of pipe that is slipped over the shaft, and rests against the opposite wall. Another method of removing a pulley is illustrated in Fig. 166, in which the piece of shaft holding the pulley is mounted on an incline, with the lower end resting against a solid wall, while another piece of old

shafting is used against the hub of the pulley, as a ram. In order to prevent damaging the pulley hub,

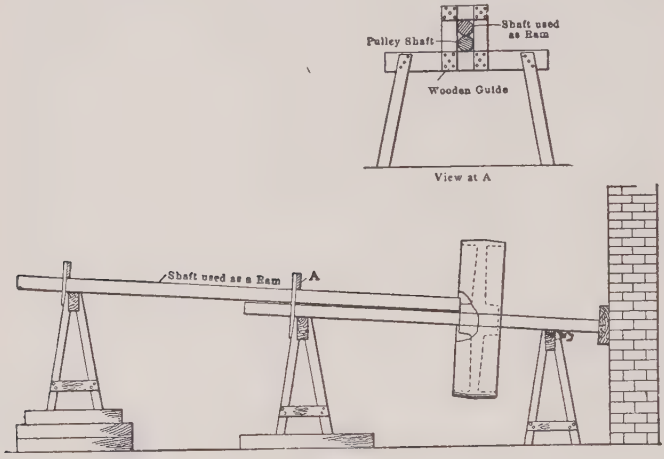


FIG. 166.

and also to have its force applied to the best advantage, the ram should be used in a direct line with the direction

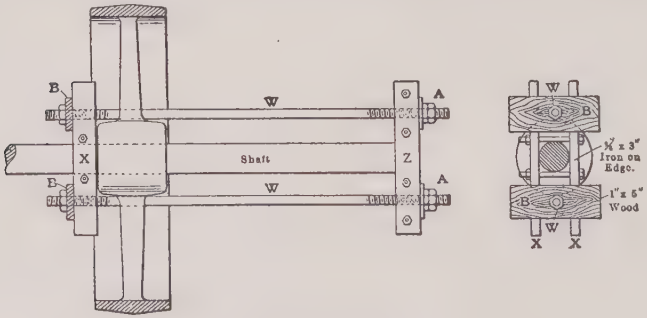


FIG. 167.

of removal. Fig. 166, is self-explanatory. A very convenient method that may be applied to any size pulley provided the bolts W W are large enough, is shown in

Fig. 167. The bolts W W must have long threads cut on them. Pressure is brought to bear on the hub by pulling up on the nuts A, A, each of which should be drawn a little at a time in order to keep the strain equalized between the two bolts. In case the pulley comes very hard, it may be assisted when the strain is on the bolts, by blows from a sledge hammer delivered

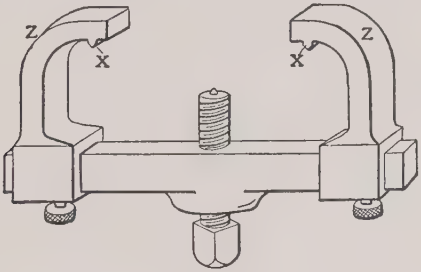


FIG. 168.

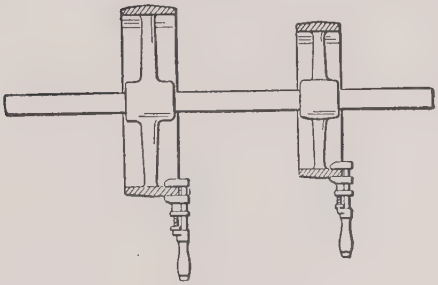


FIG. 169.

on the block x. This method can be used only in cases where the end of the shaft can be reached as shown in the illustration.

For removing pulleys from electric generators, and motors, the device shown in Fig. 168 is very convenient provided the pulleys are near the end of the shaft, and they generally are with those machines. The arms z, z,

may be adjusted so as to take hold of the hub, or the arms of the pulley, while the screw applied to the center of the shaft brings the pressure to bear.

After a pulley is once loosened, a good way to run it off the shaft is to apply a monkey wrench to the rim, and keep turning the pulley around, while the shaft is held from turning by using another monkey wrench on the rim of one of the tight pulleys. This method is illustrated in Fig. 169.

Shear in Set Screws.—Fig. 170 shows the comparative holding power of set-screws and keys, and assum-

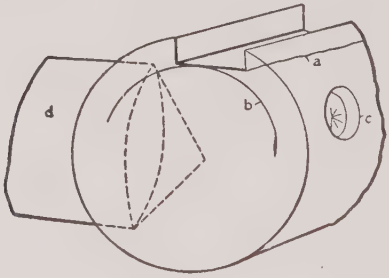


FIG. 170.
HOLDING POWER OF SET SCREWS.

ing it to show a piece of 2-inch shaft (1 15/16 inches), the key will be made ½ inch wide and it is cut into the shaft ⅛ inch. Assuming the torsional strain to be great enough to shear the shaft along the broken line a, which, in a hub 4 inches long on a 2-inch shaft, would give a section for the shear of about $\frac{1}{4} \times 4$ inches $= 1$ square inch of metal. But it is impossible for the metal to shear at the line a. Even should the piece be loose above the break line the shaft could not turn around in the hub because of the wedge-like action of the key which rides on the flat surface of the key-seat, and the entire

section of shaft along the line b must shear off before the hub can revolve on the shaft.

Thus it is the key which must be sheared off to allow the shaft to revolve, since it has been shown that the shaft cannot be sheared so the key will not hold. The key has a section along the line of shear of $\frac{1}{2} \times 4$ inches $= 2$ square inches of steel, corresponding to a set-screw 1 3/5 inches in diameter, shown at d.

WATER POWER AND ITS UTILIZATION.

Nature is very lavish in her ways of supplying energy to be utilized as power in various forms for the welfare and advancement of the human race, and in none does she excel more than in the tremendous and practically inexhaustible supply of energy that is stored in water seeking its level.

That never-ending round of circulation from the sea to the clouds, and from thence to the tops of the mountains and hills, and back again to the sea, which water is compelled by the sun's influence to keep up, will continue to be a source of power when coal shall be forgotten. As just one instance of the vast amount of power available from this source, it is said that if all the daily output of coal in the world could be used in making steam to drive pumps, it would hardly suffice to pump back the water flowing down Niagara River.

The efforts being put forth by the United States Government for the conservation of its forests and watercourses cannot be too highly commended, and should receive the hearty indorsement of all.

Water Wheel Setting.—The development of the modern turbine water wheel, with its draft tube has wrought a wonderful change in methods of wheel setting so far as

the millwright is concerned. Whereas, it was once the height of the millwright's ambition to be able to frame and erect a deck flume in which water could be confined under 10 to 40 feet head, and not a drop of water found leaking from the flume, nowadays the wooden flume, the framed penstock and the wooden wheel case are things of the past and are seldom seen, much less constructed. The boiler maker and the concrete worker do about all the water-wheel setting now, and it is done to stay. The old-time wooden construction used to last about seven years before it came time to repair it, and at ten years the whole thing had to be replaced with new material.

The open flume, the decked flume and the penstock have all been replaced by steel and cement so thoroughly that wooden flume making is an obsolete branch of millwrighting.

Modern methods have reduced the setting of a turbine wheel from a task of several weeks for a considerable force of skilled men, to the work of a few days for a foreman, a gang of laborers, and a couple of boilermakers, the latter to rivet the wheel case to the steel penstock, and draft tube. The foundation for the wheel case was made when the foundations of the mill were constructed, and all that can be seen of it are several anchor bolts projecting from a smooth concrete floor. So exact has been the engineering work with transit and station-rod, that after the wheel case has been dropped in place, it needs nothing except being twisted around a little to coincide with the shaft and building lines and perhaps leveled up a bit with some brimstone and a steel wedge or two.

The millwright in all probability will be called upon to exercise a general supervision over the work of setting the wheel case, and also of the construction of the con-

crete flume, or steel pipe line, as the case may be, but he has the blue prints, and specifications of the manufacturers to guide him in this work, and is expected to follow these.

Shaft Supports.—In order that the highest efficiency may be obtained from water wheels it is absolutely necessary that their shafts be properly supported, as regards both boxing and timbering. These are two very important factors in the proper maintenance of a water wheel.

Penstocks.—Steel, or concrete are the best materials for penstock building, unless the mill is situated in a locality where suitable timber is abundant.

Concrete should not be used for building penstocks situated above ground, for the reason that a penstock is always under a tensile strain, and concrete members should not be subjected to such strains. In fact the millwright may be at different times called upon to construct penstocks of various kinds of material, even brick being used in some cases, the kind of material used depending upon the locality, and the surroundings.

THE TURBINE.

In view of the rapidly increasing use of the water turbine as a prime mover, it may not be out of place to devote a short space to an explanation of the principles controlling its design, and the laws governing its action. The turbine is a form of water wheel which makes use of the energy developed by water as it flows between curved vanes or channels, which, owing to their shape, so deflect, and alter the course of the water in its passage that it exerts a reactionary force, thereby causing and maintaining motion. This action is an application of Newton's second, and third laws of motion, as will be readily seen.

Referring to Fig. 171, let A B represent a vane moving in the direction indicated by the arrow, with a uniform velocity c; let O A represent the direction and magnitude, v of a thin stream of water entering at A in such a manner that there is no sudden change of velocity, or direction of flow. In order that this may be accomplished, it is required that the parallelogram, Ov Ac, must be the parallelogram of velocity of the water, A C being the tangent to the vane at A and Ov, Oc must

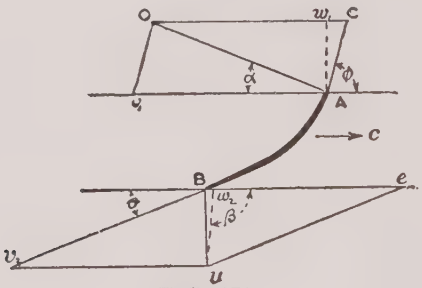


FIG. 171.

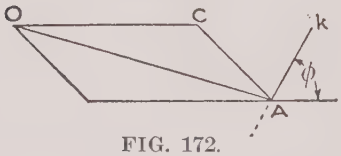


FIG. 172.

represent v, the velocity of the water relative to the vane, and c the velocity of the vane.

If, after drawing A O, O C (Fig. 172), C A were not a tangent to the vane as Ak at A, then the water would not glide along the vane without shock, but its direction and velocity would be suddenly altered at A. We must leave this latter case at present, and return to Fig. 171. A perpendicular Aw_1 upon O C gives us the tangential velocity, or velocity of whirl $Ow_1 = w_1$ of the

water at entry; the angles Φ and α may also be noticed.

Again, at the point B, where the water leaves the vane, $Bv_2 = v_2$ is the relative velocity, $Be = c$, $Bu = u$ the total velocity, and Bw_2 is the velocity of whirl at discharge, Bw_2u being a right angle, and the angles Φ and β should be noted. Let W be the weight in pounds of water passing in time t, and let P be the component parallel to the direction of motion of the vane, of the reaction between water and vane, then $\frac{W}{g}(w_1 - w_2) =$ Pt, because the momentum is changed from $\frac{Ww_1}{g}$ to $\frac{Ww_2}{g}$, in time t by a force P. Hence the work done in time t on the vane $= ct \times P$ foot pounds; $= c \times Pt = \frac{Wc}{g}(w_1 - w_2)$, a quantity independent of t. The work done per pound $= \frac{c}{g}(w_1 - w_2)$ .....(7)

If entry is to take place without shock, and certain velocities and angles are assumed, it is clear that by construction, or trigonometry the remaining velocities and angles can be found.

For example, if β θ and v_2 are assumed, then u and c are found by the parallelogram βv_2 u e. Again, if $v_f = O$ A, is known with a and c, the parallelogram Ac Ov will make it possible to find v, and θ.

However useful graphical methods may be in saving calculation, it will be found that very little more time is required and greater accuracy is obtained by the use of a slide rule and a table of sines, cosines, tangents, etc. Also, it is quite possible that calculations must be made when a drawing board is not easily obtainable, and in such cases the following equations will prove useful:

$$\left.\begin{array}{l} W_1 = V\cos\alpha \\ C - W_1 = V_1\cos\theta \\ C\sin\theta = V\sin(\alpha+\theta) \\ V_1\sin\theta = V\sin\alpha \end{array}\right\} \ldots(8) \qquad \left.\begin{array}{l} W_2 = U\cos\beta \\ C - W_2 = V_2\cos\theta \\ C\sin\theta = U\sin(\beta+\theta) \\ V_2\sin\theta = U\sin\beta \end{array}\right\} \ldots.(9)$$

Assuming, as before, B, θ, and v_2 known, then the fourth equation of (9) gives u, and the first will give W_2, and c may be found from the second or third. Again, if v, α and c are known w_1 is found from the first of equations (8); Φ may be calculated from the third, which may be written $c - w_1 = v\cos\theta\sin\alpha$, and v_1 may now be obtained from the second or fourth.

FIG. 173.

The equations (8) are unaltered if Φ be greater than a right angle; and the above reasoning will apply equally well when the vane moves, not in a straight line, but in a circle whose plane is perpendicular to the paper, the axis of rotation being then parallel to the paper. The student will now be able to understand the axial, or parallel flow turbine, through which the water flows in a direction parallel to the axis of rotation.

Figs. 173 and 174 are sectional elevations; the former is part of a cylindrical developed section taken through the guide vanes E and the wheel vanes G, the direction

of motion of G being to the left, while the guide vanes are fixed. The water, collected in a reservoir above, flows downwards into the guide apparatus, in which it

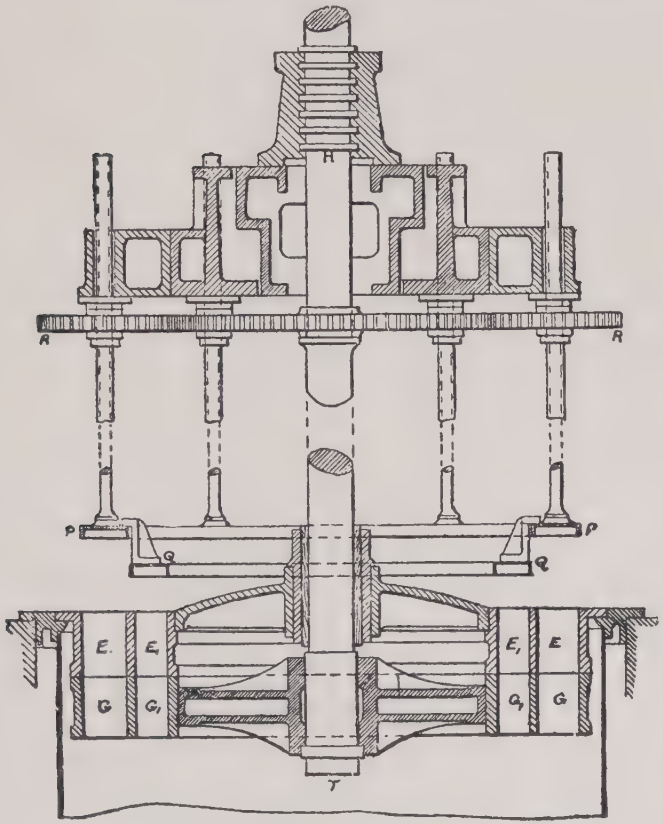


FIG. 174.

is given a forward velocity v; leaving these, it should enter the wheel G without shock, and flow out vertically, after it has imparted some of its energy to the wheel in

consequence of the change of velocity of whirl from w_1 to w_2 which should be zero, the reason for which will be explained later on. Fig. 174 is a vertical section through the axis of rotation.

The turbine of which Fig. 174 is a vertical section, is 9 feet 6 inches in diameter (outside), and is carried on a vertical wrought iron shaft 8 inches in diameter. The head of water varies from 12 feet to 16 feet, according to the condition of the river. It is capable of developing about 250 H. P. when using 200 cubic feet of water per second, with a head of 12 feet, showing an efficiency of 91.6 per cent, which is certainly remarkable, when we bear in mind that 85 per cent is considered very good. During periods of flood the quantity of the flow can be increased to about 280 cubic feet per second. The average number of revolutions is 48 per minute, giving a circumferential speed of 24 feet per second. This turbine wheel is in two parts, and is termed a double axial turbine. When the water in the river is low, there is a greater difference of level between the upper and lower water, thereby increasing the head, and during this period the outer rings, E, G, of the guide vanes and wheel are used, the inner rings, E_1, G_1, being closed. But when the water is high in the river and the fall not so great, the lower level being raised, both rings of the turbine are opened. In this manner a large volume of water with a short fall can be used, and vice versa a smaller volume with a higher fall. The center of the upper directing portion, E, of each turbine is closed by a fixed bill, in which is provided a bearing lined with strips of wood, for the turbine shaft. The shaft is carried by a compound collar bearing H, at the upper end, from which the entire weight of the turbine is suspended. This bearing is carried by a pair of cast iron girders, which

are fixed across the turbine house, resting upon the side walls.

Another bearing between H and the rings, P Q, is also given, supported by a girder. This bearing is also lined with wood strips, and above it a collar is attached to the turbine axle with clamping screws, and is adjusted so as to be just clear of the bearing in ordinary working; but when the water pressure becomes very great a slight settlement of the turbine shaft takes place, and this collar then takes a bearing, and supports part of the weight so as to prevent there being too great friction at H. The two rings, P Q, as before mentioned, regulate the supply of water. The outer ring P is suspended by six spindles, which are screwed at the top through nuts in the spur wheels R. These are geared together by a large center wheel so that they can be all turned together by gearing from a hand-wheel (not shown in the figure), so that P can be raised or lowered to adjust the opening for the passage of the water to the outer ring. Q is raised by screwing up the outer ring until it catches projecting brackets fixed to Q, which then rises with P. As usual, the power is taken from the shaft by means of a bevel wheel above H. It will be noticed that a suction tube T is used, reaching about 4½ ft. below the bottom of the wheel, so as to allow for variations in the lower level of the stream. If this were not done, head would be lost when the water is low in the river.

The theory of the action of the suction tube will be explained later on; suffice it to say here, that so long as the lower level of the water is above the bottom of the suction tube, the turbine may be placed at some height above the tail race, without any alteration of the effective head. Thus the lengths of the shaft and the six spindles may be less than they would otherwise be, the

reduction of weight of the former tending to decrease the loss by friction at H.

As has already been stated, the velocity of whirl at discharge should be zero—that is, that the water should flow out axially; or, in this case, vertically. Now u (Fig. 171) is the total velocity of outflow, but u sin β is the component of the velocity that carries the water out of the wheel. However great w_2, the other component, may be, no more water will flow through a wheel of a given size so long as u sin β is fixed. Thus, for a given wheel and given quantity of water, u sin β must have a fixed value; but u must be as small as possible, because one of the losses of head is $\frac{u^2}{2g}$, the energy in each pound of water as it leaves the wheel; but u is never less than u sin β, so that the least value of u will be when $\beta = 90$ deg., and therefore $w_2 = 0$.

From equation (7) the work done per pound

$$= \frac{c}{g} (w_1 - w_2)$$

$$= \frac{c}{g} w_1 \text{ when } w_2 = u \cos \beta.$$

Neglecting friction, the formula for ascertaining the H. P. follows: Let Q = number of cubic feet of water per second,

$$\therefore \text{ then H. P.} = \frac{Qcw_1 \times 62.5}{g \times 550} \quad \ldots\ldots\ldots\ldots (10)$$

and the maximum hydraulic efficiency is

$$\eta = \frac{c w_1}{g H} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

a quantity which varies between .65 and .9 when the turbine is running at the right speed, and none of the guide passages is closed, either wholly or partially, as this reduces the efficiency.

The ordinary parallel flow turbine is not divided into two parts as in Fig. 174, and c is then the velocity at the mean radius of the wheel vanes. Referring to Fig 174, let c, C be the velocities at the mean radii of the blades G, G_1, and w_1, W_1, the corresponding velocities of whirl q_1, Q_1, the quantities of water flowing per second through E, E_1, so that $q_1 + Q_1 = Q$ cu. ft.

$$\text{then H. P.} = \frac{62.5\ (q_1\, c w_1 + Q_1\, C W_1}{550g}$$

and hydraulic efficiency

$$\eta = \frac{q_1\ c w_1 + Q_1\, C W_1}{g\, Q H}$$

Classification of Turbines.—All turbines belong to one of two classes, called reaction and impulse turbines. In the former, when working at full power, all the guide and wheel passages are filled with water, and the turbine is said to be drowned, and the velocity of flow in one part can be determined when that in any other part is known; or, to put it mathematically, if A_1, A_2, A_3 be the cross-sections of the stream at any point, and v_1, v_2, v_3 the velocities perpendicular to those sections, then $v_1\ A_1 = v_2\ A_2 = v_3\ A_3$.

Wherever the quantity of water is large, and the fall moderate, a reaction turbine is generally used.

An impulse turbine is not drowned, the buckets are not filled, and in some cases, in which there is only partial admission, each bucket is empty during part of a revolution. Air is required in the buckets, so that the pressure may always be that of the atmosphere, ventilating apertures being made in their sides for this purpose. A suction tube cannot be used with this class of turbine, which is suitable for high falls and moderate or small quantities of water.

FIG. 175.

Thus a reaction turbine may be made for a fall of 14 feet, and 200 cubic feet of water per second; but an impulse turbine is required for a fall of 600 feet, and 20 cubic feet of water per second. In the former case an impulse turbine might be used in place of the reaction turbine, but the high number of revolutions required for the reaction turbine in the latter case, where the fall is high, makes it necessary to use an impulse turbine; for with a small quantity of water, a turbine that is filled must be of small diameter, and as the velocity of rotation at the radius at which inflow takes place is

not less than .45 $\sqrt{2g\,H}$ for a reaction turbine, and is generally greater, the number of revolutions would become inconveniently high when H is large.

Reaction and impulse turbines may be again divided into three classes: radial, axial, combined or mixed flow. In the first the water flows outwards or inwards; the

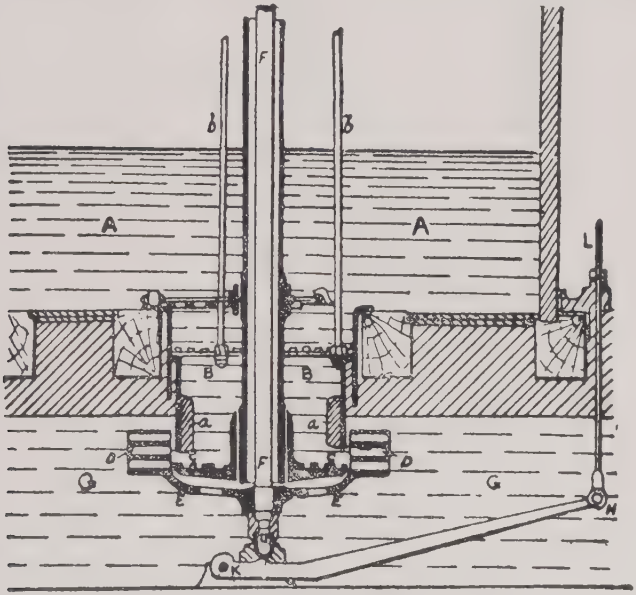


FIG. 176.

second has been already described; and in the third the water enters approximately radially, and leaves axially. The axes may be in all cases vertical or horizontal. Fig. 175 shows a section perpendicular to the axis, and through the guide apparatus and wheel of a radial outward flow or Fourneyron turbine. At B are the guide vanes and at D the wheel vanes, the arrow showing the

direction of rotation. Fig. 176 is a vertical section of this type of wheel. A is the penstock, B the supply cylinder, which consists of two concentric tubes; the upper one being fixed, while the lower slides within it like the inner tube of a telescope, and is raised or lowered by the rods b. Near the upper edge of the inner tube is a leather collar making a watertight joint between it and the outer tube. The lower part, a, of the inner tube acts as a regulating sluice for all the orifices at once. It has fixed to its internal surface wooden blocks, shaped so as to round off the turns in the course of the water towards the orifices.

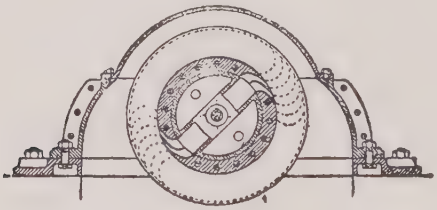


FIG. 177.

The bottom of the supply cylinder is formed by a fixed disc, C, which is supported by hanging at the lower end of a fixed vertical tube enclosing the shaft. This disc carries the guide blades. D D are the vanes of the wheel, which in this case are divided into three sets, or horizontal layers, by two intermediate crowns, or horizontal ring-shaped partitions. The object of this is to secure the filling of the passages by the stream at three different elevations of the sluice, and thus lessen the loss of efficiency which occurs when the opening of the sluice is too small. E is the disc of the wheel, F is the shaft, G the tail race. KH is a lever which supports the step, and is itself supported by fixed bearings at K, and

by the rod L, which can be raised or lowered by a screw so as to adjust the wheel to the proper level. The turbine shown in Fig. 176 is a reaction turbine, but its mechanical construction is the same if it works as an impulse turbine. In the latter case, however, the turbine is not drowned, the discharge taking place above the level of the tail race.

Fig. 177 shows an impulse outward flow turbine. The wheel is of cast iron, and the flow takes place at the opposite ends of a diameter, the two guide passages at either end being made of wrought iron. The valve which

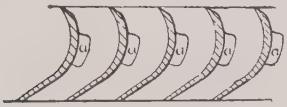


FIG. 178.

FIG. 179.

FIG. 180.

regulates the opening of these is made of gun metal, and is keyed to a spindle which can be rotated through the small angle required to close the passages, by hand or by a governor. The following are particulars of two turbines similar to the above:

	I	II
Inner diameter of wheel	11.81 in.	11.81 in.
Outer diameter of wheel	16.14 in.	14.84 in.
Number of vanes	45 cast iron	84 wrought iron
Revolutions per minute	583	928
Quantity of water per minute	4.52 cu. ft.	5.085 cu. ft.
Head of water	129 ft. 1.6 in.	131 ft. 5.19 in.
Available power per minute	36,440 ft. lbs.	53,806 ft. lbs.
Power measured on brake	16,498 ft. lbs.	28,809 ft. lbs.
Efficiency, per cent	45.27	53.5

Figs. 178 and 179 show sections through the wheel, a a being the ventilating aperatures. The great breadth

of the wheel shown in Fig. 179 will be noticed. It is so made to prevent the passages becoming filled at outflow Fig. 180, because if this occurred, the wheel would not work as an impulse turbine should, for the reason that while in reaction turbines, continuity of flow is a necessity, in impulse turbines it must not take place because

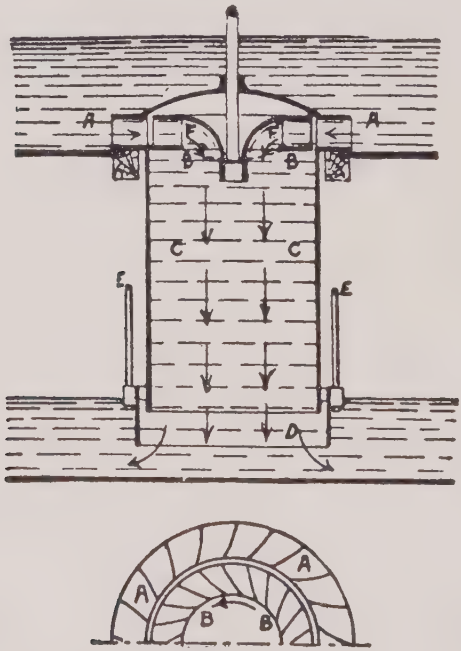


FIG. 181.

the pressure must always be that of the atmosphere, for which reason the ventilating aperatures are provided.

Fig. 181 is an outline drawing of an inward flow turbine with suction tube and cylindrical sluice at the bottom. The lower part of the figure is a sectional plan through guide apparatus and wheel, marked respectively

AA and BB. C is the suction tube, while D is the regulating sluice, by no means an economical method of regulation, because, as the passage at outflow from the sluice is diminished the water flows more slowly through the whole wheel, having also a diminished velocity of whirl at entry, together with a shock. Again, on leaving the wheel it still has some tangential velocity left, because the velocity of the water relative to the wheel is less than when the sluice is raised. The velocity of the stream passing away from D is also increased, thus creating three

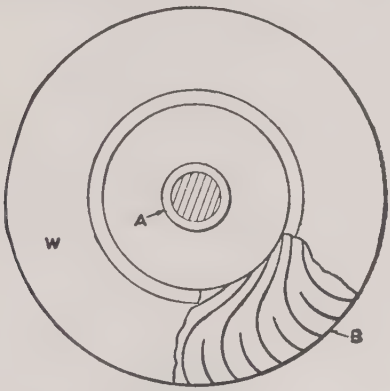


FIG. 182.

important causes of loss. The shaft is carried by a collar bearing above. The direction of flow is indicated by the arrows. The letters FF, and the dotted curves below them represent the vanes of an inward and parallel flow wheel, and do not refer to the inward flow turbine. The water is turned from a radial to an axial direction, hence the term "mixed flow" for this type.

The Thompson inward flow turbine is shown in Figs. 182, 183 and 184. Fig. 182 shows the wheel, the portion B being in section, its sides, W, are conical (Fig. 183).

The water enters at O, Fig. 184, and flows around in both directions, as shown by the arrows. Instead of the area being greatest at O and least at the side opposite O, it decreases from O uniformly around the circumference, the water flowing in the direction taken by the hands of a clock, so that no changes in direction may take place.

The guide vanes, D, are hinged at E, and can be moved by links C so that the passage to the wheel is enlarged or diminished, according as the guide vanes move

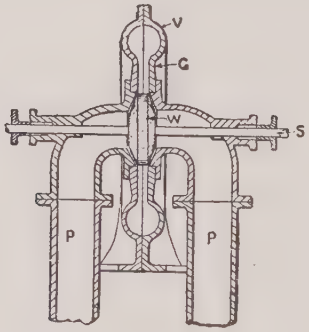


FIG. 183.

clockwise, or in the opposite direction. This wheel shows good efficiency, either at full load or light load. Alteration of the guide vane angle has been very successful with both inward and axial flow turbines, although some energy is lost by shock at entry with the gate partly closed, and as the volume of water flowing through the wheel is then less, the relative velocity of discharge is less. The water finally leaves the wheel in directions parallel to the axis, and passes out of the casing down the draft tubes, P P.

The Suction Tube.—As has already been stated, the

suction tube enables a reaction turbine to be placed in a higher and consequently more convenient position than it otherwise would be, shortening the shaft and regulating rods, and sometimes, as in the case of the Victor turbine, allowing bevel wheels to be dispensed with, and pivot friction exchanged for journal friction, which lat-

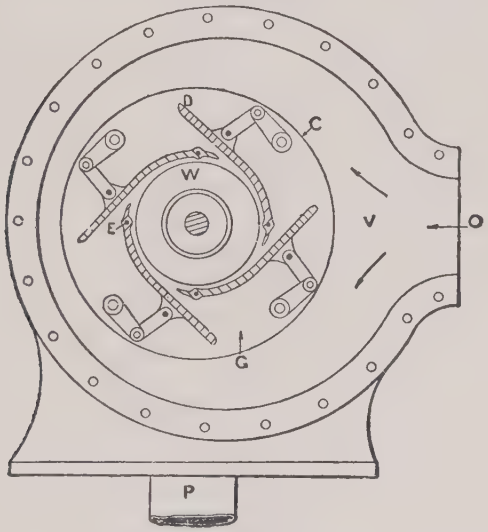


FIG. 184.

ter has been shown by the latest experiments to be less than the former.

Fig. 185 shows a tube, A B, in which is a piston P, C D being the upper level and E F the lower lever of the water. Now, suppose the atmospheric pressure is replaced by the pressure of two columns of water, A C and K B; then the pressure on P is evidently unaltered. But if we express the pressure on P in feet of water, the pressure on its upper surface is A P, while that on

its under surface is P K; the effective pressure is therefore A K, and

$$\begin{aligned} A\,K &= A\,C + C\,K \\ &= K\,B + C\,K \\ &= B\,C \end{aligned}$$

so that as long as P is below K where K B is the height of the water barometer, the head is unaltered.

Now, with a given turbine with a given resistance to overcome, and with a given head, no matter how that head is obtained, whether by water pressure or atmospheric pressure, a certain number of revolutions will be

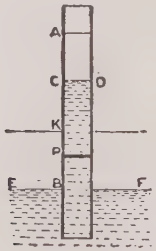


FIG. 185.

obtained, and a certain quantity of water will be used. This is evident, because the state of affairs at the turbine is unchanged, for the pressure at every point is unchanged, as shown above; and if the forces and the sphere in which they act are unchanged, the motions will also be unchanged.

This, then, explains how a turbine with a draft or suction tube can be placed above the tail race. The height is theoretically limited to 34 ft., but practically to much less. More accurately, if v_3 be the velocity of flow from the suction tube, theory would give

$$h_3 = 34 - \frac{v_3^2}{2g}$$

where h_3 is the above height.

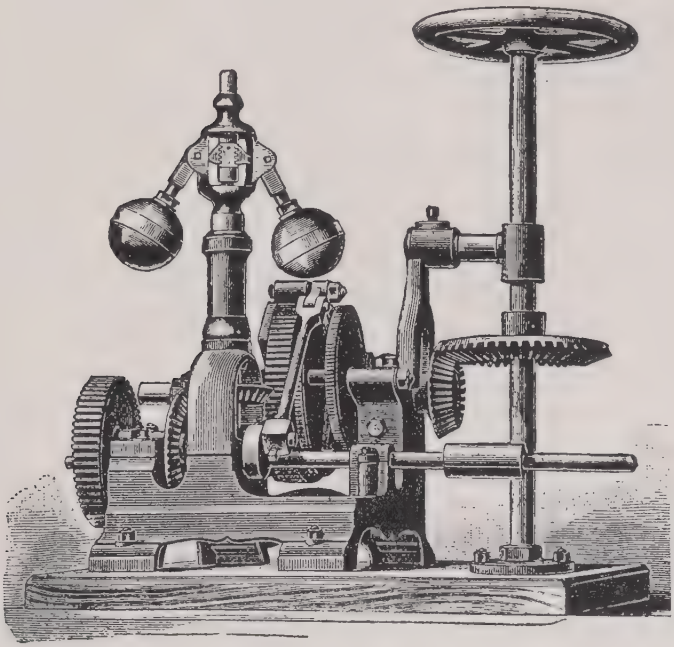


FIG. 186.

In practice, however, in the above equation the following quantities should be substituted instead of 34 for the corresponding diameters of suction tube. This table is given by Meissner.

Diameter of tube in feet.	h_3
.49	31.16
.98	29.52
1.6	27.88
2.3	27.88
3.3	26.24
4.9	19.68
6.5	14.76
8.2	13.77
9.8	12.46
11.5	11.15
13	9.84

Turbine Governors.—All water wheel governors are of the indirect acting class; that is, they enable the wheel to close or open the sluice by its own power, the governor itself not being powerful enough to perform the work directly. Figs. 186 and 187 show two views of the Snow governor as applied to the Victor turbine. Fig. 186 is a perspective view, and Fig. 187 is an outline drawing, showing the principle of its action. Referring to Fig. 186, it will be noticed that there are two shafts, one vertical, and the other horizontal. The former turns the pinion which gears with the spur segment on the sluice, and the latter drives the governor. Upon this horizontal shaft at its right end is a pulley, not shown in the figure, which is driven by belt from the turbine shaft, while at the left end there is a pinion which is concealed by the spur wheel at the extreme left of the figure, which spur wheel the pinion drives, and this, by means of two bevel wheels, drives the governor balls. The arms of the governor have teeth on their inner ends, which are in gear with the central spindle, so that according as the balls

fall or rise, the spindle rises or falls. Fig. 187 shows the connection between this spindle and the pawl shifter. B C D is a lever with a fulcrum at C. D G is a link connecting it with the sector, which is called the pawl shifter, and which is pivoted at H. Referring again to Fig. 186, a small crank will be noticed at the right of the governor base, and upon the same shaft that carries the spur wheel at the left of the figure. This crank gives a reciprocating

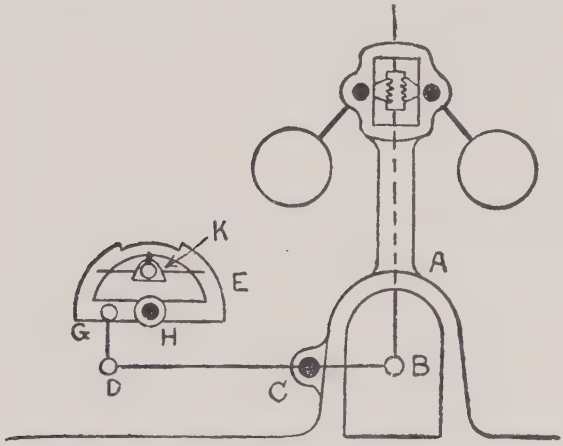


FIG. 187.

motion to the short connecting rod, at the upper end of which are two pawls, termed the closing and the hoisting pawls, because one closes and the other opens the sluice when they are allowed to gear with the ratchet wheel that lies just at the left of the upper end of the connecting rod. The sector, or pawl shifter (Fig. 187), part of which is just visible to the left of the ratchet wheel in Fig. 186, would, if its upper edge were circular, prevent the pawls gearing with the ratchet wheel; but the depression at the center of the circumference, Fig. 187,

enables them to fall into gear when required. For, if the balls rise, B is lowered and D and G are raised, so that the depression on the pawl shifter allows the closing pawl to gear with the ratchet wheel. If the balls fall, motion in a reverse direction takes place, and the hoisting pawl gears with the ratchet wheel. This ratchet wheel turns the vertical shaft by means of the two bevel

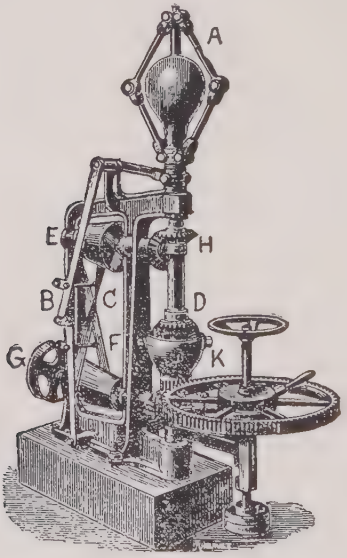


FIG. 188.

wheels shown at the right of Fig. 186, the larger of which is on the vertical shaft. In order to prevent overwinding, there is an arrangement of reducing gear between the ratchet wheel and bevel pinion, which brings into position a stop similar to the sector, or pawl shifter, which will allow the pawls to slide freely along its top, and prevents them engaging with the ratchet wheel, and thus prevents further motion of the sluice. In order to

regulate the speed, there is a horizontal bar on the pawl shifter, upon which slides a weight, K, Fig. 187. If this is pushed to the right, it will tend to raise D and lower B, and lift the governor balls, so that in order that the pawl shifter may be in its middle position, with neither pawl in gear, the speed of the wheel must be less. For a similar reason, if is desired to increase the speed, K should be shifted to the left. This governor is

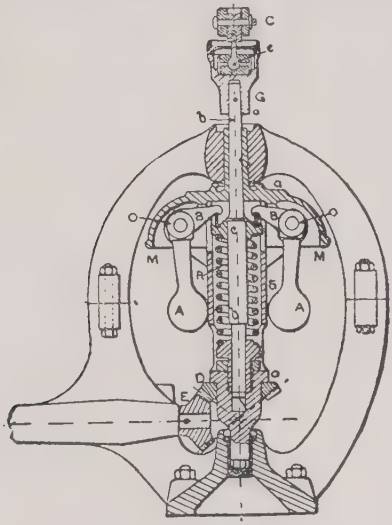


FIG. 189.

said to be very rapid in its action. Hett's governor is shown in Fig. 188. Its action depends mainly upon the differential motion of three bevel wheels.

In this governor the initiative is given by a governor of the Porter type A, fitted with heavier balls than usual. As this rises or falls it moves a belt shifter, B, which changes the position of the cross belt C on the cone pulleys E and F. Thus when the governor sleeve is in mid

position, the belt is on the center of the cones and E is driven at the same speed, but in the opposite direction to F, which receives its motion from the pulley G. When

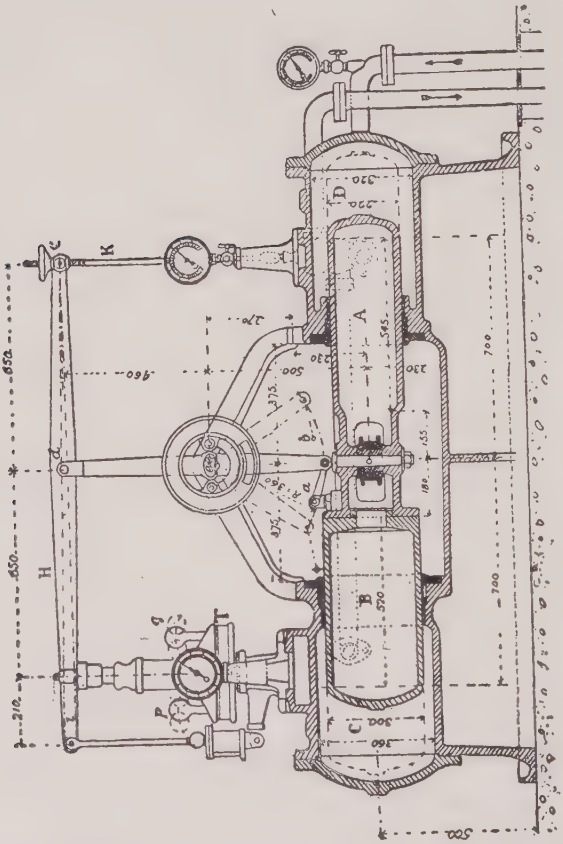


FIG. 190.

the sleeve falls the belt is shifted to the left, so that E is driven faster than F, and when the sleeve rises the reverse takes place. The lower cone shaft drives the governor spindle by means of miter bevel wheels. The

upper cone shaft drives a sleeve H D, which is loose on the governor spindle, and carries the upper bevel wheel of differential motion D at its lower end. The lower pinion of D, being attached to the governor spindle, revolves with it. As the gearing is the same, it is clear that when the governor is running at its proper speed, so that the belt is on the middle of the cones, the upper and lower bevel wheels of the differential motion run at

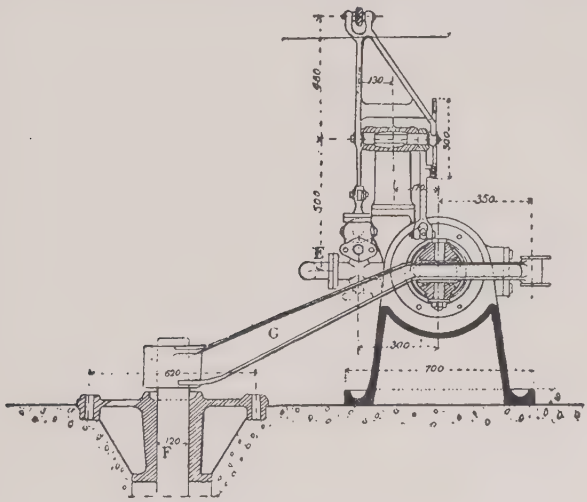


FIG. 191.

the same speed in opposite directions, and the two other bevel wheels in the cup K rotate on their axes, but do not give motion to cup K. When the upper bevel wheel runs at a different speed to that of the lower one, the cup K rotates, and moves the pinion at its base, which gears with the spur wheel on the hand-wheel spindle of the starting gear. To connect the motion of the hand-wheel spindle, a spring clutch is employed, which can

be instantly thrown in or out of gear by giving a quarter of a turn to the disc, shown immediately above the spur wheel. The clutches are thrown out when stopping by hand, and will slip if the running speed is not reached when the gate is fully opened. One great advantage of this arrangement is that the speed of regulation increases in proportion to the amount of irregularity to be corrected.

Fig. 189 shows a form of governor constructed by the Vevey Iron Works. The balls, A A, are carried by the

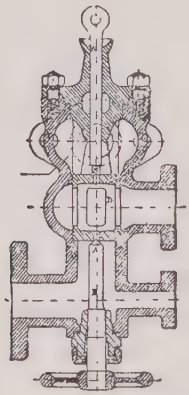


FIG. 192.

bell cranks, B B, having knife edges, O O, which are carried by arms, M M, that are keyed to the governor shaft, a a. The ends, B B, of the bell cranks rest upon a sleeve, c, which carries the movable central rod, b b. The small pieces of steel between B B and the sleeve should be noticed. They reduce the friction considerably, as they are free to turn through a small angle.

The part C acts on the valve of the servo-motor, whose duty it is to move the sluice or gate of the turbine. The servo-motor of the Bellegarde turbines is illustrated in

Figs. 190 and 191. The water under pressure acts upon a differential piston, A B, connected at the center to a lever, G, which turns the shaft, F, controlling the sluice. D is permanently connected to water under pressure, while C is put in connection by the governor, T, with water under pressure, or with the discharge. In the first case the differential piston moves to the right, and in the second case it moves to the left. The former motion closes, and the latter opens the sluice. The distribution of water under pressure to C is effected by a valve S, shown in detail in Fig. 192. This valve rests on the upper orifice of cylinder D, while the other orifice at the bottom is throttled more or less by the point of the regulating needle, A B, Fig. 192. The valve S is suspended by the rod K at the point e of the lever H, which is pivoted at d, and supported by the index i of the governor, T. When the governor moves i, it produces at the same time a motion of valve S. As the lower orifice of cylinder D, Fig. 192, is connected to the water under pressure, while the upper orifice is in connection with the discharge to the atmosphere, and the space between these two orifices is in communication with the large cylinder, C, it will be seen that when the index of the governor rises, and the valve S is lowered, the upper orifice is closed, and water under pressure flows into cylinder C through the lower orifice. When, on the other hand, the governor index descends, valve S rises and opens fully the upper orifice, which has a section almost double that of the lower, so that the pressure falls in C and the water escapes from it. If, now, the valve is midway between these two positions, so that the two orifices present the same section to the water, the pressure in the cylinder C remains about midway between the initial pressure of the water and the pressures on the differen-

tial piston A B are balanced so that it does not move. There is, however, this disadvantage, that there is an

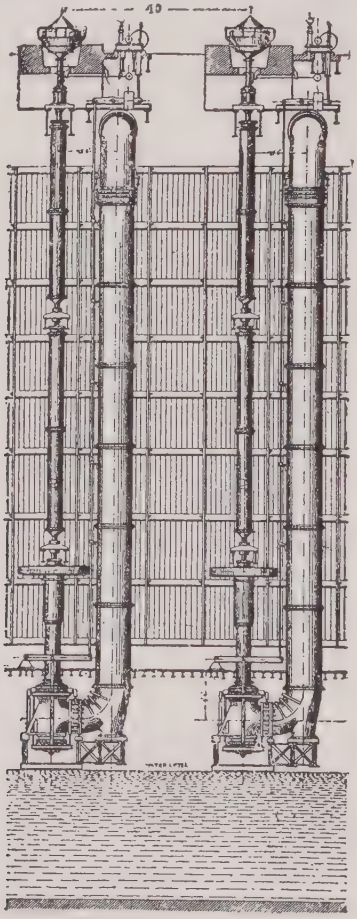


FIG. 193.

almost continual flow of water through m. On the other hand, there are two great advantages. First, the small

valve S is only rarely forced to the end of its stroke by the governor. Generally, it is only partially moved,

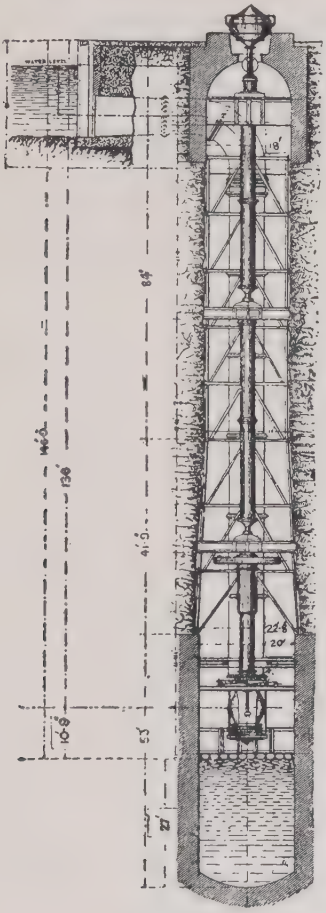


FIG. 194.

and from this results a speed of gate opening, or closing more or less great.

It is only when there is a considerable change of load that the valve is pushed down by the governor, and the velocity of the motion of the piston, A B, reaches its maximum value. A small change of load, on the other hand, produces a change in the opening and closing of the sluice proportional to the magnitude of the change of load. The small valve is acted on by the liquid with a variable force. When it descends, the upward force on it increases from zero to several pounds, and this gives the governor stability. The pistons must be so proportioned that the resistance of the sluice is easily overcome, and the orifices in D, Fig. 192, must also be of such a size that the rapidity with which the sluice is closed may be such as is desirable. Generally, orifices of from 0.24 in. to .39 in. will suffice.

So far, apparently, the governor only puts in and out of gear the apparatus for closing the sluice, and for that purpose only a very small motion of the index is necessary; but in practice the piston, A B, and consequently the sluice, takes up, when the wheel is running steadily and the load is constant, a position determined by that of the index of the governor. The point of rotation d of the lever H is not really fixed; it is connected to the piston, A B, in such a manner that it rises or falls proportionally to the motion of this piston. It results from this, that when the piston is at rest, and consequently the valve S occupies its mean position as well as the point e, the index i of the governor is obliged to fix itself in the position corresponding to that of the point d, and, reciprocally, the point d and the piston, A B, are obliged to follow the movement of the index i. But when the wheel speeds are in a state of oscillation the index i acts freely upon e independently of the motion of d. Regarding the motions of d and A B, the piston, A B,

moves by the connecting rod a b the crank b c, which carries at C an eccentric, whose rotation raises or lowers the point d by means of the rod d c. The displacement of d is approximately proportional to the movement of A B.

Turbines in Use at Niagara Falls.—Figs. 193, 194 and 195 show the general arrangement of these turbines, which were designed by Foesch and Piccard of Geneva, and built by I. P. Morris & Company of Philadelphia. Each turbine operates under an average head of 136 feet, and consists of two wheels to each shaft.

The speed is 250 R.P.M., with a volume of water equal

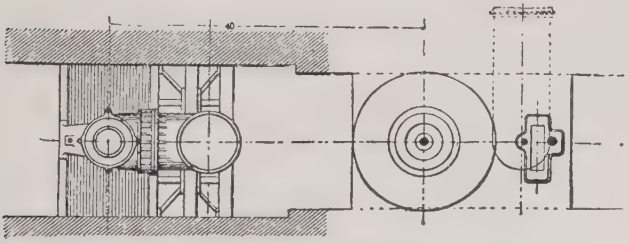


FIG. 195.

to 430 cubic feet per second, and each turbine develops 5,000 horsepower. The shaft is vertical and drives an electric generator direct connected at the top. The wheels are of bronze, and are shown in Figs. 196 and 197, while the lower wheel is shown to a larger scale in Figs. 198 and 199. They are reaction wheels, although the fact that they discharge into the atmosphere might lead one to suppose at first that they were impulse turbines.

There are 36 guide, and 32 wheel vanes. A steel pipe 90 inches in diameter conducts the water from the canal (Fig. 200) to a chamber between the two guide wheels.

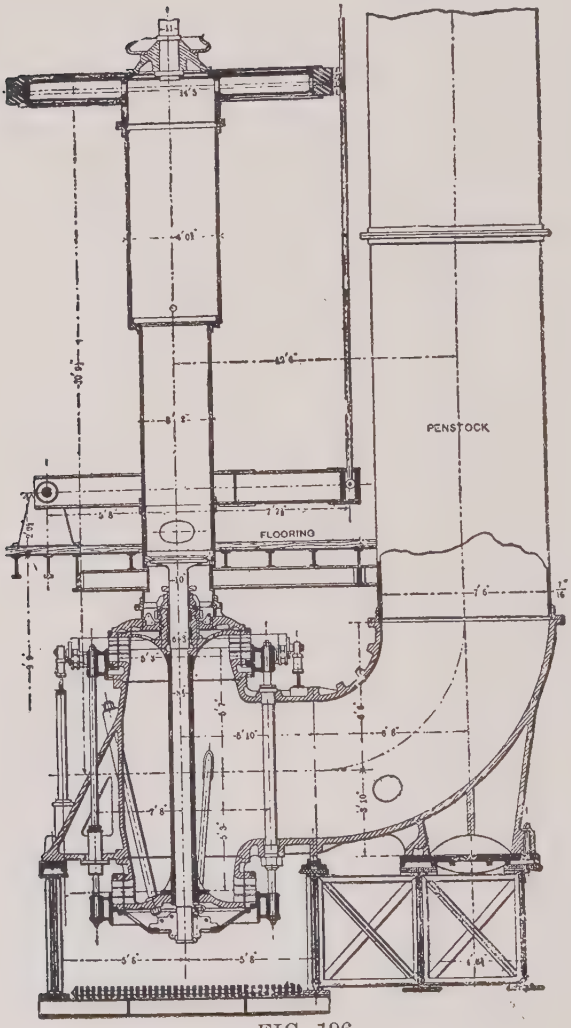


FIG. 196.

The main shaft is made of steel tubes 38 inches in diameter, and 11-inch solid shafts are interposed at the points where the guide bearings are placed. This method of construction at once lightens the weight, and increases the rigidity of the shaft, and also reduces the number of guides necessary. A heavy fly-wheel of forged iron, weighing 10 tons, and having a diameter of 14 ft. 6 in. is attached to the shaft, as shown in Fig. 196. The

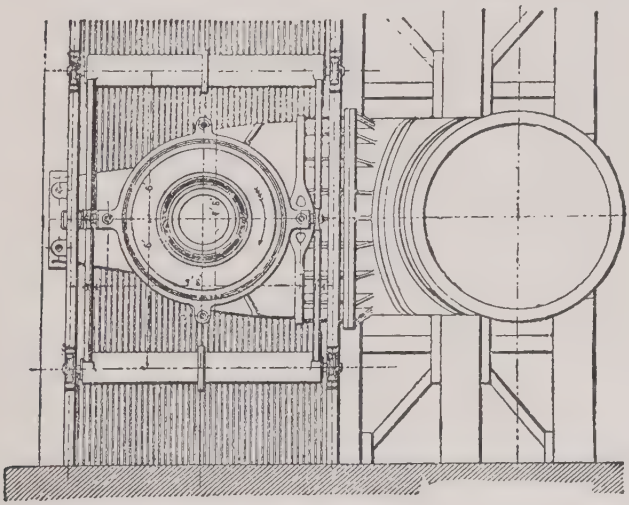


FIG. 197.

peripheral speed of this wheel is 11,000 ft. per minute. The upper bearing of the shaft is entirely relieved of its load when the turbine is running, due to the pressure of the water acting upon the under surface of the upper wheel. The speed is regulated by ring sluices, not between the wheel and guides, but outside the wheel. The sluice is raised or lowered by means of a system of rods and levers (clearly shown in the figures), which

are under the control of a very sensitive governor of the indirect acting class. The speed varies between a maximum of ½ per cent at the normal rate of working, and about 3 per cent if the load is increased or diminished suddenly by one quarter. This sensitiveness is very necessary for an electric installation.

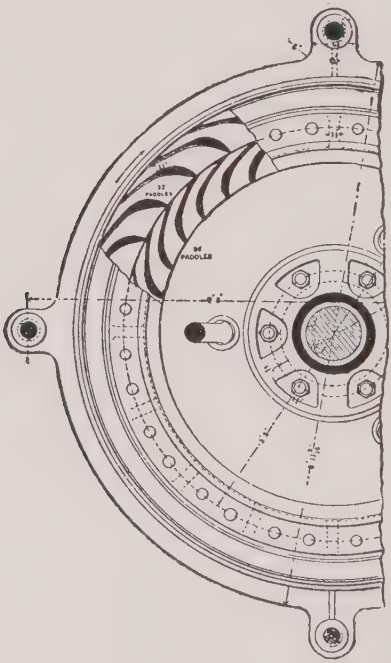


FIG. 198.

A tunnel has been made to carry off the water from the wheels. It is 6,700 ft. long, with a section of 490 square feet, and a hydraulic gradient of about 7 ft. in 1,000, and discharges its water into the river again below the Falls, and near the Suspension Bridge. The velocity of flow is given by Professor Forbes as 25 ft. per second,

which we believe to be unprecedented. Ordinarily, about 8 ft. per second is considered as the extreme limit in brick-lined sewers; indeed, a leading waterworks engineer allows only about 3 ft. as a maximum in lined channels. The lining here is made of 16 in. of hard brick, this having been found necessary because the rock passed through disintegrated or crumbled quickly when exposed to the air. The brick and hydraulic cement used in the construction of the lining had united to form a

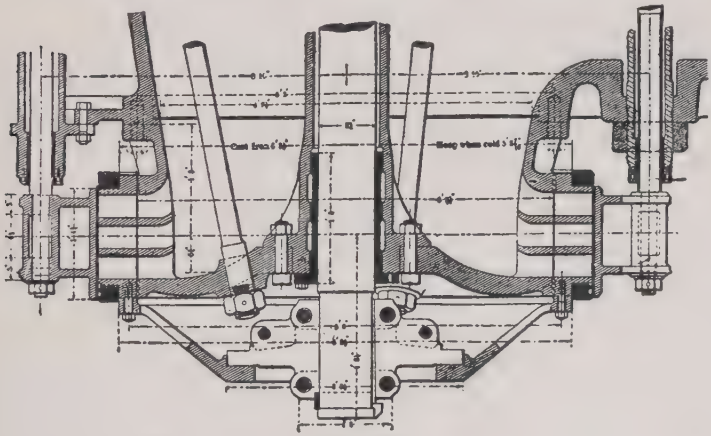


FIG. 199.

mass as solid as rock in some specimens taken from the tunnel.

The "Poncelet" Water Wheel.—Fig. 201 shows a type of water wheel known as the "Poncelet" that is adapted to falls not greater than 5½ feet. The stream flows under a sluice, and enters the buckets of the wheel without shock, owing to the curvature of the vanes. The stream then rises up in the vane on account of its relative velocity, v, and falls again, flowing out radially. The sluice is inclined at an angle of about 50 degrees. Some

of the head must be lost if the wheel does not work in back water, as Fig. 201 shows the water leaving the wheel at some little height above the tail race. Theoretically, the arrangement should be as shown in Fig. 202, where D E is a horizontal line and the angle D E C is bisected by the line C A, so that the fall of the water, after it has attained its greatest height in the wheel is the same as its rise.

Neglecting friction, $v_2 = v_1$, $\theta = \eta - \Phi$, and $c_1 = c_2$.

$$\text{and} \quad \left.\begin{array}{l} c_1 = v_2 \cos \theta \\ \dfrac{c_1}{\sin(\alpha + \Phi)} = \dfrac{v_1}{\sin \alpha} \end{array}\right\} \ldots\ldots\ldots (1)$$

$$\sin(\alpha + \Phi) = \sin \alpha \cos \theta = -\sin \alpha \cos \Phi \ldots\ldots (2)$$

$$\left.\begin{array}{rl} 2 \sin \alpha \cos \Phi = & -\cos \alpha \sin \Phi \\ 2 \tan \alpha = & -\tan \Phi \end{array}\right\} \ldots\ldots\ldots (3)$$

Generally, D C A = about 15°.

$$\alpha = 15° \ldots\ldots\ldots\ldots\ldots (4)$$

An efficiency of about 70 per cent has been obtained with this wheel, but this is rather high for calculation, 66 per cent being more suitable.

The Pelton Water Wheel.—This wheel, also known as the tangential water wheel is shown in Fig. 203. It is made up of a number of double buckets, fixed to the circumference of a wheel. Water projected from a nozzle strikes the buckets at their centers, and is deflected equally to both sides.

Let v = velocity of water projected from nozzle
c = velocity of center of buckets
v_1 = relative velocity of inflow
v_2 = velocity of outflow.

A section of a bucket is shown in Fig. 204, and it will be seen that shock will occur at inflow owing to shape of the central partition, which theoretically should be like a knife edge, but as this is impracticable, it will be ignored for the present, and assuming that the coefficient of resistance referred to v_2 is F, the following equations appear:

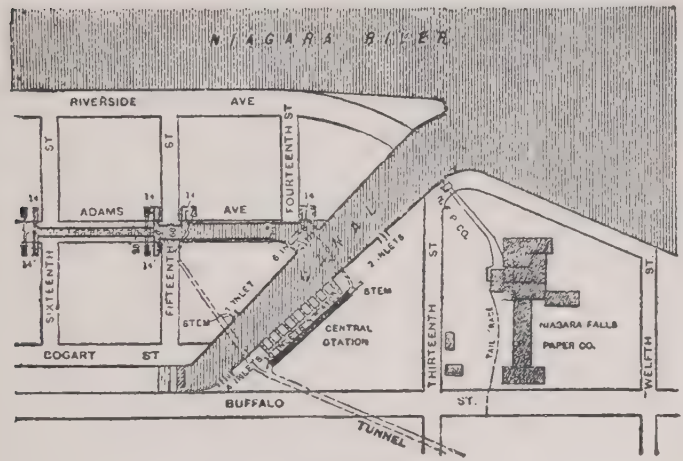


FIG. 200.
CANAL—NIAGARA FALLS POWER STATION.

$$\text{Work done by wheel} = \frac{c\,(w_1 - w_2}{g},$$

in which w_1 and w_2 are the tangential velocities of the water at inflow, and outflow,

and
$$w_1 = v$$
$$w_2 = c - v_2$$
$$v_1 = v - c$$

$$\frac{v_2^2}{2g}(1+F) = \frac{v_1^2}{2g} = \frac{(v-c)^2}{2g}$$

The total wasted energy is:

$$\frac{Fv_2^2 + W_2^2}{2g},$$

which must be a minimum for maximum efficiency. We therefore have

$$(1)\ \frac{F}{I+F}(v-c)^2 + (c-v_2)^2 \text{ as a minimum}$$

$$= \frac{F}{I+F}(v^2 + c^2 - 2vc) + c^2 + \frac{(v-c)^2}{I+F} - \frac{2c\ (v-c)}{\sqrt{1+F}}$$

$$(2)\ \frac{F}{1+F}(2c-2v) + 2c - \frac{2\ (v-c)}{1+F} - \frac{2v-c}{\sqrt{1+F}} + \frac{2c}{\sqrt{1+F}} = 0$$

$$(3)\ c = \frac{v}{2} \text{ whatever the value of F.}$$

Figs. 205 and 206 show improved types of the Pelton wheel, in which the streams of water strike the buckets alternately on each side of the continuous dividing edge

as shown in the cuts, thus securing a steadier motion. The impulses are divided more regularly on the wheel, as each bucket passes the point of the nozzle and

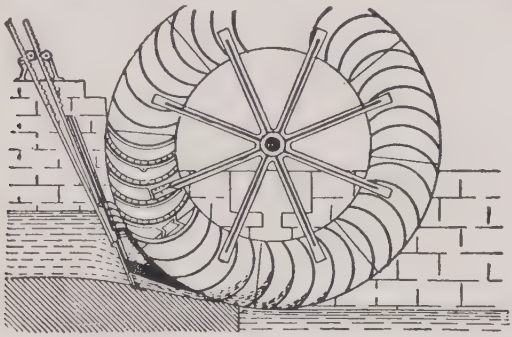


FIG. 201.

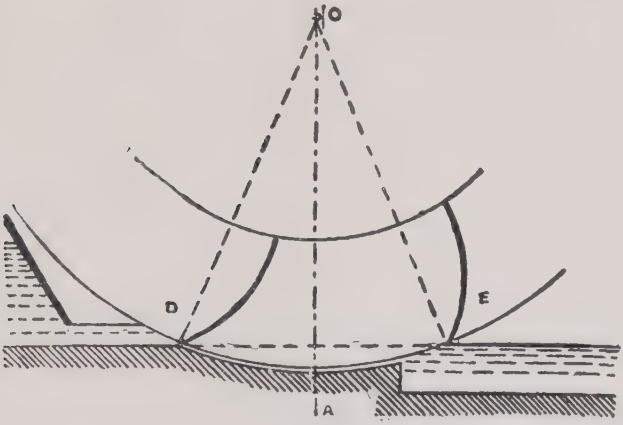


FIG. 202.

catches its share of the water. These buckets are cast solidly upon each side of the circular dividing ridge, and upon the face, or rim of the wheel on each side of this central division. This circular ridge being also

angular, and curved as it approaches the center, gives to the interior of the buckets a symmetrical, and effective curve.

The further claim is made by the builders that this arrangement of buckets, and form of construction secure great strength, rigidity and stability, and the buckets

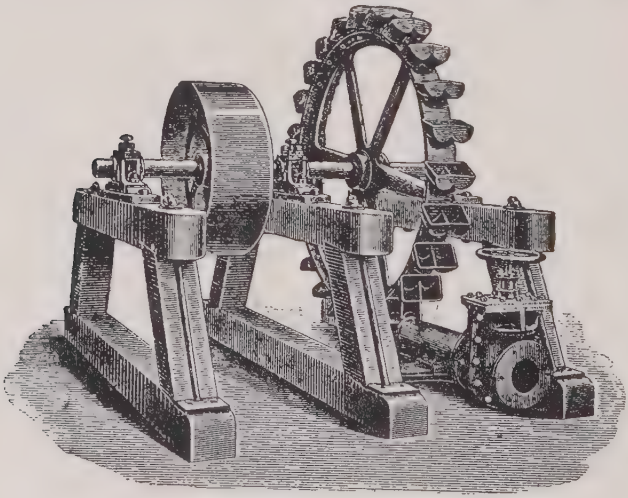


FIG. 203.

FIG. 204.

are not subject to the difficulty of becoming loose. In this type of wheel several jets are used. The method of mounting is such as to permit the moving of these jets, and thereby varying the angle of inclination at which the water may be projected on the buckets. Either of the nozzles may be removed, and others of different size, or bore substituted, or any of them may be capped over, and one or several, or all used as may be desired.

SETTING HYDRAULIC RAMS.

Hydraulic rams are water-raising appliances in a class by themselves. The shock that is commonly noticed on quickly closing a fullway cock and suddenly stopping the flow of water in long lengths of pipe is the power employed in the hydraulic ram.

FIG. 205.

The hydraulic ram is a machine of very simple construction, but it does not seem to be understood as it should be by those who may have the erection and installing of it. It consists of a body, or trunk, and an air-vessel, in which are inserted three valves, namely: (1) The pulse, or foot valve at end of the trunk; (2) the re-

taining, or ascension, valve, in the air-vessel; and (3) the snifting valve, in the neck of the air-vessel, immediately under the ascension valve. The office of the snifting valve is to supply air to the air-vessel when the ram

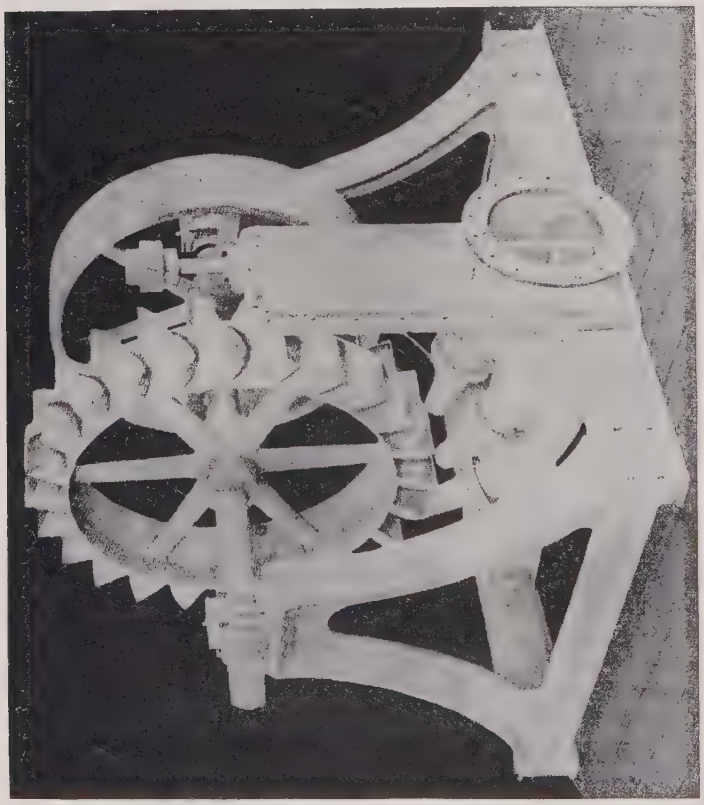


FIG. 206.

is working. This, though seemingly the most unimportant valve in the ram, is most necessary to the successful working of the machine, as without it the ram would soon cease working, owing to the air having been ex-

hausted, as the air-vessel would have become waterlogged. Air escaping from the air-vessel with the water causes the severe shocks and noise in badly constructed rams that are not fitted with this simple valve.

The hydraulic ram is a machine that utilizes the momentum of a stream of water having a slight fall, in such a way as to elevate a portion of that water to a greater height. This machine is self-acting; and when once set going, it will go on working day and night for a long period without stopping, provided that the supply

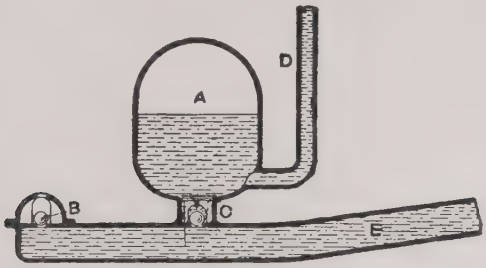


FIG. 207.
SECTIONAL ELEVATION OF HYDRAULIC RAM.

of water is sufficient and that the ram has been properly constructed, and fitted up.

The hydraulic ram will force water with a fall of from 18 in. to 100 ft. to an elevation of 15 ft. to 1,000 ft. from almost any distance, from a few score of yards to four or five miles, or more if necessary. For example, assuming that 100 gal. of water falling 10 ft. would elevate 10 gal. to a height of say 80 ft., as 100 gal. falling 5 ft. will elevate 1 gal. to a height of 300 ft., a hydraulic ram will raise water from 300 gal. up to 500,000 gal. per day of twenty-four hours if required.

The following example will give some idea of the power exerted at the end of a long pipe when the flow

is suddenly stopped: A pipe flowing full bore with a velocity of 25 ft. per second is equal to a head of 10 ft. If the pipe is 2 in. diameter and 150 ft. long, the contents will be $\frac{2^2 \times 62.5}{144} \times .7854 \times 150 = 204.5$ lbs., which multiplied by $10 = 2{,}045$ ft.-lbs. of energy.

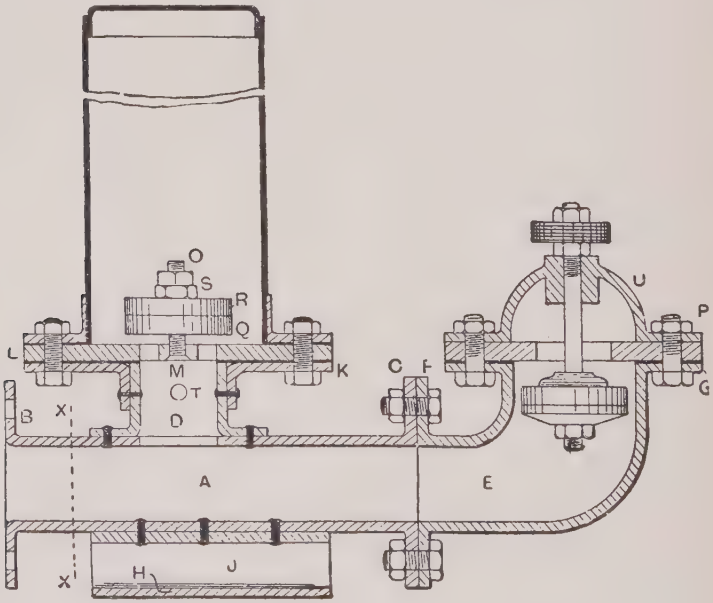


FIG. 208.
VERTICAL SECTION OF COPPER HYDRAULIC RAM.

A diagrammatic section of a hydraulic ram is presented by Fig. 207. A is an air vessel, B and C ball valves, D the delivery pipe, and E the supply pipe. Above valve B is an opening, and the water running down from a small fall at E, passes this outlet until the

velocity is sufficient to close B. This of course suddenly stops the stream, and the outlet valve is forced open, owing to the great increase of pressure in the ram. Through C the water passes into A and up the delivery pipe D. This releases the pressure, and valves B and C drop to their seats, and the operation is repeated. In

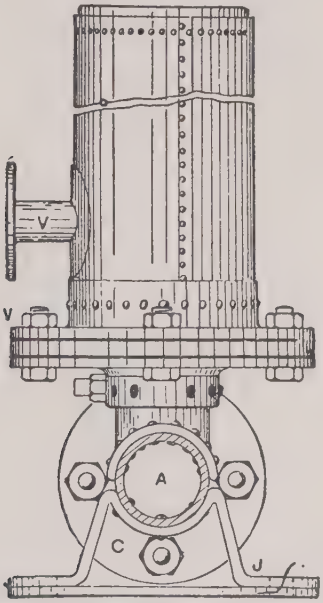


FIG. 209.
END ELEVATION OF COPPER HYDRAULIC RAM.

some cases an ordinary lift, or flap valve, weighted to slightly exceed the static pressure of the supply stream, is placed between E and C. Obviously, a small portion only of the supply water from a small fall is delivered to a greater height, and the average efficiency of the ram is probably not more than 50 per cent.

Figs. 208 and 209 show sectional, and end elevations of a 1-inch ram in copper sheet or tube adapted for springs or streams supplying from 90 gals., to 300 gals. per hour, and raising approximately 5 to 20 gals. per hour to a height of 80 feet. These illustrations are so proportioned however, that a ram may be constructed from them to suit any fall of the drive pipe, and any quantity of water, by simply increasing the dimensions according to the varying conditions of the stream, or head.

The volume of water raised by a hydraulic ram varies according to the ratio of the fall to the height that the

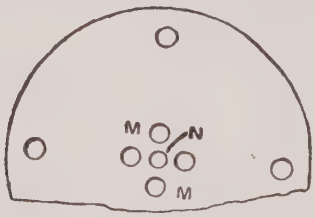


FIG. 210.
PLAN OF DELIVERY VALVE SEAT FOR HYDRAULIC RAM.

water is to be lifted, while the effective capacity is materially affected by the length of the drive and delivery pipes, as well as by the relative fall and lift. A ram will work on a fall as low as 2 ft. 6 in., and the quantity of water raised will be in proportion to the fall. Generally speaking, a proportion of 1 in 10 of fall to lift gives the greatest efficiency. As there are no two streams whose conditions are exactly alike, it is obvious that a ram which will work well on one stream, will not work satisfactorily on another. Therefore, to obtain the highest efficiency, the ram must be designed to suit the conditions of the stream on which it is to work. These conditions are ascertained by surveying the site, and mak-

ing other observations. The data required are as follows:

(1) the minimum volume of flow per minute supplied by the stream in summer; (2), the horizontal distance from the stream to the proposed site of the ram; (3), the vertical height that the water is to be lifted above the ram; (4), the vertical height obtainable from the source of supply; (5), the horizontal distance from the ram to the end of the delivery pipe. The method of making these observations will be described later on. The ram illustrated in Fig. 208 and 209, may be constructed of sheet-copper, or drawn tube, the latter being the most satisfactory. In order to withstand the pressure, the thickness should be not less than 3/32 in. for low falls of 5 feet, and a short drive pipe.

For falls above this height, and for long drive pipes, the thickness should be $\frac{1}{8}$ in. Fig. 208 is a section of the ram, and Fig. 209, an end elevation, with the connecting flange cut away on line xx (Fig. 208). The body A is $5\frac{3}{8}$ in. long when finished, and to allow for the flanges B and C, the sheet or tube should be $7\frac{1}{8}$ in. in length. The width of the sheet (if used) will need to be sufficient to make a tube of 1-in. internal diameter, and to allow for a $\frac{3}{8}$-in. lap for the joint. Small rivets are used for making the joint, which is afterwards soldered. The edge of the lap on the inside should be bevelled off, and care should be taken that the heads of the rivets do not project much into the ram. The flanges are worked over as shown, to $2\frac{3}{4}$ in. in diameter. At $1\frac{3}{4}$ in. from one end a hole 1 in. in diameter is cut, the center of the hole being $2\frac{1}{4}$ in. from the end; over this hole is riveted the T-piece D, communicating with the delivery valve.

The T-piece is of the same diameter as the body of the ram, and when finished it is 1 inch high. The sheet

or tube will need to be cut sufficiently long to allow of a flange ¾ in. wide, being worked on the end resting on the body of the ram. The T-piece is riveted to the body as shown, and it is also soldered so as to make a water tight joint.

To avoid having rivet heads on the inner surface of the ram, the T-piece and body are bolted together with

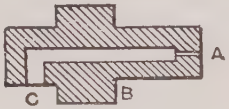


FIG. 211.

SECTION OF SNIFTING VALVE FOR HYDRAULIC RAM.

FIG. 212.

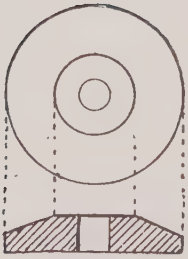


FIG. 213.

FIG. 214.

DETAILS OF PULSE VALVE OF HYDRAULIC RAM.

⅛ inch brass wire, having a fine thread cut on it, the holes in the flange of the T-piece and body being tapped for this thread. When all the studs are screwed up, they are cut off nearly flush, say to about 1/64 in. of the inside and outside surface of the copper, by filing, after which a 1-inch iron bar is inserted in the tube, and the brass studs are riveted down by a few blows on the

outside with a hammer. The joint is then sweated together with solder, and if there are any burrs left on the inside ends of the studs, they should be smoothed off with a file, to prevent obstruction to the passage of the water. The elbow E, Fig. 208, is of copper tube 1 inch internal diameter. For rams working on even low falls the thickness should be not less than 1⅛ in. so that the bend may be worked out to the larger diameter as shown, the elbow being enlarged where the pulse valve works to an internal diameter of 1⅝ in. The tube should be of sufficient length to allow of the flanges F and G being worked over.

Flange F which is bolted to the body, is 2¾ in. in diameter, and flange G is 3⅜ in. in diameter. The approximate length of tube required is from 7½ in. to 8 in. Holes for ¼ in. bolts are drilled through the flanges as shown. The bed plate H and J, Fig. 208, is made of two pieces of sheet brass ⅛ in. thick. The bottom piece H is 3¼ x 4 in., and to this is bolted and sweated, the cradle J, Fig. 209, made out of a sheet about 3¼ x 6¾ in., shaped as shown, to fit the body of the ram.

The bed plate and body are riveted or bolted, and sweated in the same manner as is the T-piece. The flange K, Fig. 208, is ⅛ in. in thickness by 4 inches in diameter. From the center the metal is worked out at right angles to form a ring with an internal diameter of 1¼ in., this ring fitting over the T-piece, to which it is secured by riveting.

To this flange is bolted the circular valve seat L, a plan of which is shown in Fig. 210. This valve seat is made of a brass plate 4 in. in diameter by ¼ in. thick. The four water ways M, M, are ¼ in. in diameter, and the center hole N is drilled and tapped for a ¼-in. stud O, Fig. 208, on which the delivery valve works. The

stud after being screwed in, is riveted over on the under side as shown.

The water passages are drilled with their centers ⅜

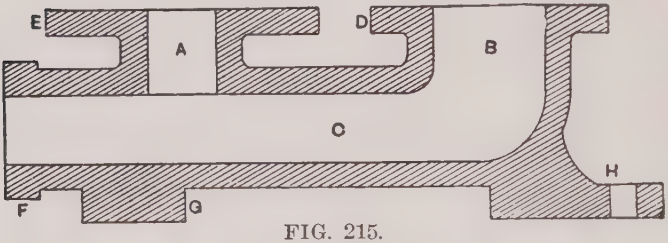


FIG. 215.

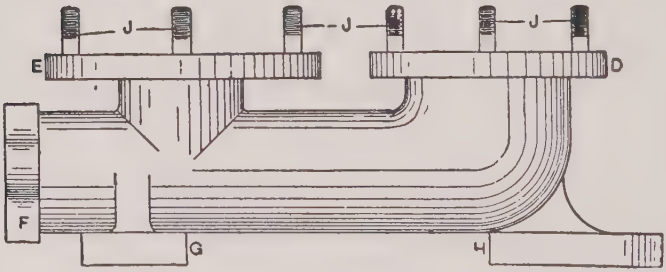


FIG. 216.

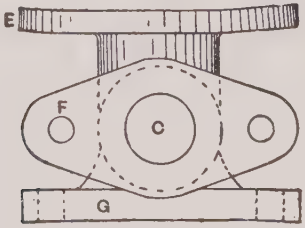


FIG. 217.

SECTION, SIDE, AND END ELEVATIONS OF CAST-IRON HYDRAULIC RAM BODY.

in. from the center of the stud. The pulse valve seat P is a brass plate 3⅜ in. in diameter by ¼ in. thick. A water way 1 in. in diameter is drilled in the center, this

being the diameter of the bore of the ram. Four holes for ¼-in. bolts are drilled in both valve seats. For larger rams the number and size of the bolts should be increased, the spacing being 2 in. from center to center. The delivery valve consists of a rubber disc Q, and a brass disc R, both being 1⅜ in. in diameter by ¼ in. thick.

Through the center of each, a hole is made for the stud O, which should fit loosely. The brass disc acts as a weight to keep the rubber disc in position, and the lift of the valve is adjusted by the nuts S. The delivery valves of small rams should not have more than ⅛-in. lift. For larger sizes the lift is proportionately greater.

A small split pin is put through the stud above the nuts, to prevent their working loose. The snifting valve or air inlet, is shown at T, Fig. 208, and in section in Fig. 211. It consists of a brass plug having a ⅛-in. hole drilled through its center, tapering to a very fine hole at the inner end A, where it should not be larger than would allow a very fine needle to pass through it. The enlarged part B is hexagonal for the wrench, and at C, a hole is drilled at right angles to the first, to guide the water downward, and prevent splashing, as a small quantity will squirt out, when the pulse valve closes. The snifting valve is screwed into the T-piece through the lap of the flange K (see Fig. 209) in place of one of the rivets. The object of this valve is to allow a little air to pass into the ram at each beat of the air vessel.

The details of the pulse valve are shown separately at Figs. 212, 213 and 214. The spindle (Fig. 212) of the valve is a ¼-in. diameter mild steel rod 3½ in. long with a thread cut at each end for a length of 1¼ in. A brass disc (Fig. 213), 1⅜ in. in diameter by ¼ in. thick, tapered off at its outer edge as shown, forms the

under support of a rubber disc, 1⅜ in. in diameter by ¼ in. thick. The upper brass disc (Fig. 214) is ⅞ in. in diameter by ¼ in. thick. The central hole in the brass disc is tapped to fit the thread of the valve spindle.

In putting the valve together the upper brass disc is first screwed up tight on the spindle with the flat side downwards (see Fig. 208); over this is placed the rubber disc; and over this is screwed the supporting brass disc, which must not be screwed up so tight as to press the rubber out of shape. Below this disc is a lock-nut; and further to prevent the disc and nut working loose while the ram is working, a small hole is drilled through the spindle for a small split pin.

At the other end of the spindle are two nuts and brass washers for adjusting length of stroke, and weight of the pulse valve according to the length and fall of the drive pipe, and the height to which the water is to be raised. The guide sleeve U for the pulse valve spindle is made of a strip of sheet brass ¾ in. wide by ¼ in. thick, bent as shown, the central portion being ¾ in. long and ½ in. in diameter. This sleeve must be fixed perfectly square with the lugs to allow the valve to rest squarely on its seat. The valves are faced with rubber in order to reduce noise and wear.

The air chamber, Fig. 209, has an internal diameter of 2¼ in. and is made of sheet copper 1/16 in. thick, riveted and soldered together. The flanged copper pipe v near the bottom is the outlet to the discharge pipe, the diameter depending upon the length of the delivery pipe. With a long delivery pipe, the diameter for this sized ram should be not less than ½ in., but with a short delivery ⅜ in. would be sufficient.

It is unnecessary to go into details regarding the construction of this portion of the ram, as the cut ex-

plains itself. Fig. 215 is a sectional elevation of the body in cast iron. Fig. 216 shows a side elevation, and Fig. 217 an end elevation of the body. A is the water way to the delivery valve and air chamber. B is the passage leading to the pulse valve, C waterway in the body, and D flange to which pulse valve seat and guide valve are bolted. E is the flange to which are bolted the delivery valve seat, and air chamber, F flange for connecting drive pipe, and G back lugs for bolting ram to the foundation, there being a bolt on each side of the ram. H is a lug cast at the end of the elbow for bolting the front to the foundation, and J short studs ¼ in. in diameter used for bolting the valve seats and the other fittings to the ram. In order to obtain the highest degree of efficiency from a hydraulic ram water-raising plant, the horizontal length of the drive pipe should equal the vertical height that the water is to be raised, and the diameter should be such as to discharge three times the available quantity. The diameter of the delivery pipe should be such as not to add more pressure on the ram than is due to a head of 3 feet.

The area of the pulse valve water way should equal that of the drive pipe, and the area of the delivery valve water way should be as large as possible, to permit a free passage for the water, with a short lift of the valve, thus preventing the water flowing back past the valve, causing slip. The capacity of the air chamber in cu. in. should not be less than the cubic capacity of the delivery pipe.

Flow of Water.—When gauging streams with a very small quantity of water flowing, a triangular notched weir should be used, as this gives more accurate results with small quantities than a rectangular weir.

The formulæ for making hydraulic ram calculations are:

$$(1)\quad g = \frac{G \times f}{L} \times 0.65$$

$$(2)\quad G = \frac{g \times L}{f} \times \frac{1}{0.65}$$

$$(3)\quad f = \frac{g \times L}{G} \times \frac{1}{0.65}$$

where G = gallons of water flowing down the stream to work the ram; g = gallons of water raised; L = lift, or height the water is to be raised above the ram; f = the vertical fall of the feed water; and 0.65 the approximate efficiency of the ram described in this article.

To show the working of these formulæ, a stream is taken as an example, having an available fall of 5 ft., with 2.819 gal. flowing per minute. It is required to raise the water to a tank 50 ft. above the ram, the length of the delivery pipe being 60 ft. What quantity of water would be raised to the tank in 24 hours? Applying formula 1, the quantity raised, assuming that there is practically no frictional loss in the delivery pipe, will be $\frac{2.819 \times 5}{50} \times 0.65 = .1832$ gal. per minute, or $.1832 \times 60 \times 24 = 263.8$ gal., say 260 gal. in 24 hours, with 4,060 gal. passing through the ram in 24 hours. Now, applying formula 2, what quantity of water will be required to raise .1832 gal. to a height of 50 ft. above the ram, the supply water having a fall of 5 ft.? The quan-

tity will be $\frac{.1832 \times 50}{.5} \times \frac{1}{0.65} = 2.819$ gal. per minute.

The working of these two examples will show the use of the formulæ, and from the data obtained by them, the dimensions of any ram may be calculated.

In calculating the required diameter of a ram to pass a given quantity of water through a given length of drive pipe, on a given fall, the quantity that is to pass through the ram is multiplied by 3, for the reason that the water in the drive pipe flows only about 1/3 of the time towards the ram, and the diameter must be calculated for the maximum flow at any moment.

Taking the above quantity of 2.819 gals. per minute on a fall of 5 feet, and assuming that the water is to be raised 50 feet above the ram, the length of the drive pipe will be $50 \div 3 = 16\ 2/3$ yards, or say 17 yds. long, and the maximum flow will be $2.819 \times 3 = 8.457$ gals. The required diameter of the ram, will by this rule, be

$\sqrt[5]{\frac{8.457^2 \times 17}{5}} \div 3 = 1$ in. in diameter. The maximum quantity that would pass through a ram of a given diameter, length of drive pipe, and fall is found by the rule for ascertaining the discharge of a pipe in gals., and dividing the result by 3. The rule follows: multiply area of pipe in square inches by velocity of water in feet per minute and divide the product by 144; the result is number of cu. feet discharged per minute. Multiplying this by 62.5 will give number of pounds, which divided by 7.5, will give number of gals.

The diameter of the delivery pipe for a given fric-

tional loss is found by the same rule as that for the diameter of the ram; but instead of dividing by the available head, the head to be lost in friction is taken. In this case the quantity flowing through the pipe is not multiplied by 3, as the water has a practically constant flow.

For short delivery pipes the head due to friction may be ignored, but for lengths above 150 feet the head that will be lost must be added to the actual height the water is to be raised, and the diameter must be calculated for this loss. The capacity of the delivery pipe is found by multiplying the area of the pipe in sq. inches by its

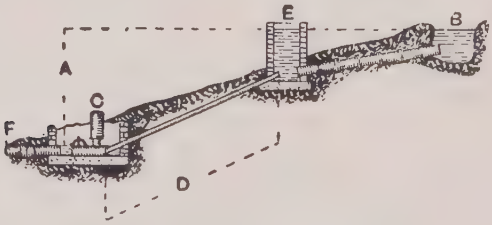


FIG. 218.
METHOD OF SETTING HYDRAULIC RAM.

length in inches. The result will be the capacity of the pipe in cubic inches, and the air vessel will have to be of the same capacity. In the case of long delivery pipes, to prevent the air vessel being of inconvenient length, it should be made baloon-shaped. By these simple rules it is possible to calculate the dimensions of a ram that would work satisfactorily with water from any stream.

Fig. 218 shows the method of fixing the hydraulic ram, A being the vertical difference in feet between the water surface at the source of supply B and the ram C; D is the drive pipe, which is of the same length as the vertical height to which the water is to be raised. As

will be seen, the fall is an even gradient from the source of supply to the ram, and the distance between the ram and the source is greater than the length of the drive pipe; it will therefore be necessary to build the small brickwork tank E, the top of which must be a little higher than the surface of the water at the source. This tank is used to prevent the bursting of the pipes when the water is suddenly checked by the closing of the pulse valve, for with long pipes the pressure at the moment when the valve closes would be greater than the pipes and joints could withstand. The ram is bolted to a solid concrete foundation, and should be placed in a small house as shown. The level of the pulse must be at such

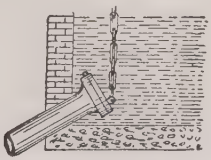


FIG. 219.
FLAP VALVE ON DRIVE PUMP.

a height that it is never covered by the tail-water, which is drained into a ditch at F.

The drive pipe should be laid at an even gradient throughout. With small rams up to 2 inches in diameter, this pipe may be of wrought iron or steel with threaded couplings, but for larger rams cast-iron flange pipes should be used of a thickness sufficient to withstand the pressure without bursting. All the joints must be water tight, and when wrought iron pipes are used, their ends should butt together in the couplings. To prevent the contraction of the column of water at its entry to the drive pipe, the end in the tank should be fitted with a trumpet mouth and flax valve, controlled by a chain, as shown in Fig. 219.

The tank and source of supply are connected by ordinary socketed drain pipe, having a diameter 2 to 3 times that of the drive pipe. The joints in the supply pipe are made with cement. A copper strainer should be fitted to the inlet end of the pipe to prevent debris from entering and choking it. The opening in this strainer should be of greater area than the pipe. As a further protection against the strainer becoming blocked by weeds and rushes collecting around it, a wire netting may be placed around it and secured to posts driven into the bottom of the stream near the pipe. The posts are driven in about 1 ft. away from the pipe on each side,

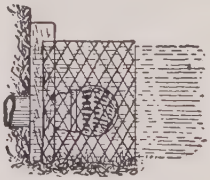


FIG. 220.
DETAIL OF STRAINER.

and the netting fixed to allow a space of 1 ft. between the netting and the strainer.

The ram having been fixed in position and the drive pipe and the supply pipe to the tank laid, an attempt can be made to start the ram. The starting is effected by pressing down the pulse valve so as to let the water discharge and then allow it to rise, continuing to work it in this way for a few strokes by hand, when it should continue to beat by itself if the adjustment of the valve is suitable for the conditions of the water supply and the height the water is to be raised. But more probably it will be found that the ram will not work before it is adjusted. The adjustment is effected by the two nuts and the brass washers at the end of the valve spindle.

The nuts are for adjusting the length of the beat, and the washers are for adjusting the weight of the pulse valve by adding to or taking away. The length of the beat, and weight of the valve depend upon the length of the drive pipe, and head of water. The stroke or beat of the valve should be as long as possible for efficient working.

On a slow and long stroke the ram will use and lift more water than on a fast and short one. The number of strokes is reduced and the length increased by unscrewing the nuts, and the number of strokes is increased and the length decreased by screwing them up. The weight of the pulse valve should be about the same as that of the water in the drive pipe when it is at rest. To adjust the valve to obtain the best results will occupy considerable time, as many trials will have to be made, by reducing or increasing, as required, the weight of the valve, and the number and length of the stroke, until a satisfactory result is obtained.

Stoppage of the ram after it has been working for a considerable time may be due to any of the following causes: The area of the drive pipe may be reduced or roughened by rust, or a deposit of organic matter may have formed on the inner surface of the pipe. In this case, although the head may be sufficient to dash the valve on to its seating, the force of the recoil may not be sufficient to allow the valve to open ready for another stroke. To remedy this, a bundle of wire should be dragged through the drive pipe several times, by means of a length of rope, after removing the ram from the pipe. Leaky joints in the drive pipe may also prevent the water recoiling sufficiently to open the pulse valve. The variation in level of the head water is a very common occurrence where the supply is intermittent. The

level of supply being lowered, the air is carried in with the water, causing the valve to chatter for a short time, and ultimately to remain closed, though if the valve is a heavy one it will remain open. The air vessel may be water-logged, owing to the snifting valve becoming inoperative, through being covered by the tail-water; or the snifting valve may be choked, causing the resistance in the air vessel to be so great that the delivery valve will not open, thus stopping the delivery of water, though the pulse valve will continue to beat, and everything will appear to be in perfect working order. The recoil of the drive water may be insufficient to allow the pulse valve to open. With a slow lift, the absence of air in the air vessel may sometimes be detected by the water being delivered in a succession of spurts accompanied by concussion in the pipes. To remedy this, shut the flap valve on the end of the drive pipe in the tank, remove and empty the air vessel, and see that the snifting valve is in good order. On replacing the air vessel, the ram should again deliver water properly.

To stop the ram, the pulse valve is held up for a few seconds against its seating, when it will cease to beat; but to stop the ram for examination and repairs, the head or flap valve in the tank must be closed.

Efficiency.—In tests, the highest efficiency obtained from a ram was 74.9 per cent, but this varies according to the ratio of lift and fall, being greater at low than at high ratios. Clark gives the following percentages of efficiency for ratios varying from 4 to 26:

Ratio of lift to fall.....	4	6	8	10	12	14	16	18	20	22	24	26
Efficiency per cent.....	72	61	52	44	37	31	25	19	14	9	4	0

Thus with 10 feet head and 40 feet lift, the ram would show an efficiency of 72 per cent. But with 10 feet head and 260 feet lift, the ram could just raise water to that

height, but it would be unable to deliver any; hence its work at that ratio would have no efficiency.

CONSTRUCTION OF DAMS.

In the construction of a dam across a stream of water, especially if the bottom is of an alluvial nature, the main problem is to successfully guard against the water finding its way underneath the dam instead of over it. Of course reinforced concrete construction when properly put in is the best method of solving the problem of a foundation for a dam, but it is not always possible to use this material, owing to objections on the part of the

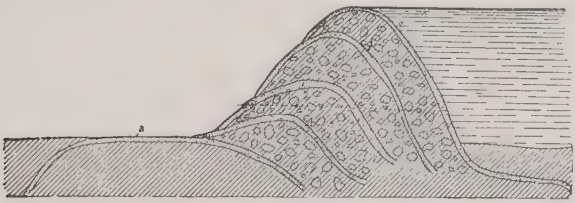


FIG. 221.
WIRE CLOTH DAM FOR ALLUVIAL STREAMS.

owners regarding its high cost. But when it can be used, the undermining of the structure may be prevented by a properly constructed concrete blanket extending for some distance above and below the dam. In case concrete is not used, the next best method of getting a foundation is by means of heavy timbers termed mud-sills which are imbedded in the bottom as deeply as possible. Upon this the dam may be constructed of any available material, and by the use of the wire cloth apron illustrated in Fig. 221, a dam may be constructed that will withstand the strain of any freshet that may come. Reference to Fig. 221 will explain the method of utilizing wire cloth for this purpose, a representing the apron extended as far

down the stream as local conditions make it necessary. Both upper and lower ends of this wire cloth apron should be imbedded in the bottom of the stream as deeply as it is possible to excavate. Having once secured the apron a to the bottom, a pile of any material at hand such as dirt, ashes, brush, logs and stones, is placed upon it. When earth filling is used, one part of cement should be mixed with it, the cement and earth being put in place dry. These are mixed by the handling while being placed, but should not be wetted at any time, that part of the work being left to the river, and natural moistures.

If during the operation of filling, the number of shovels full to a wheel-barrow load is counted a few times, the matter of mixing the earth and cement is easily solved, the proportion being one shovel full of cement to 99 shovels full of earth, the cement being sifted into the barrows while they are being filled with earth. The filling is tamped as firmly as possible along line b, Fig. 221, and the angle of this line should not exceed the angle of repose of the material used, which will be 30 degrees if sand is used, while if clay-like material is used this angle may approach 60 degrees.

Having thus secured layer a, b, in its place, another wire-cloth blanket may be placed, the wire simply laid down upon a thin coating of concrete, and another thin coating spread over the wire-cloth which is strengthened by lapping the strips, according to necessity. The concrete had best be made of aggregate not over ¾ inch in diameter in order that the blanket may be kept down to about 2 inches in thickness.

Proceed in like manner to place filling b c, ramming and smoothing as before; then lay down blanket c, and so continue until the required hight is reached. The last blanket e is to be extended up the stream as far as neces-

sary to prevent the water from getting under the dam. In some cases it may be necessary to extend this layer 100 feet or more up stream. The ends of the blankets which come against the river-bed are extended as far into the earth as it is possible to excavate. The upstream end of each blanket may be extended as far as is thought necessary for the safety of the structure. Dams built in this way will stand a great deal of wear. and give good service.

THE ELECTRIC MOTOR IN MILLWRIGHTING.

Considered from a mechanical standpoint only, there is little choice in point of reliability between a direct-current motor, and an alternating-current induction motor. The direct-current motor armature has the mechanically undesirable commutator and brushes and, usually, a larger proportion of insulating material than the alternating-current motor which, however, labors under the inherent mechanical disadvantages of a very small air-gap or clearance for the rotating element.

The particular method of application selected for any type of motor has more direct bearing on the durability and life of the motor than is commonly supposed. This can be appreciated when it is remembered that the revolving armature of any type of motor is composed of a great number of parts, coils of wire, etc., held together by bands or wedges, and depending to a greater or less extent on fibrous or cotton insulating materials of little mechanical strength and, therefore, more susceptible to injury from jars and vibration than other forms of revolving machinery. It is important, therefore, when considering a motor application, to choose a method of connection which will tend to absorb or nullify these ef-

fects unless the motor has been designed with a special aim to meeting these conditions, as in railway motors, special mill motors, etc.

Belt Connection.—Wherever conditions will allow, belt drive should be used, as the belt forms the most flexible connection between motor and load and is the best-known absorbent of mechanical shocks due to load variations. For ideal conditions the belt speed should be from 4,000 to 5,000 feet per minute.

In general the load per inch width for single, or light double belts should not greatly exceed 40 pounds for belt speeds under 4,000 feet per minute and should not exceed 32 pounds for speeds from 4,000 to 6,000 feet, which is the practical limit of working. The total working pull on belt may be found from the following equation:

$$\text{Pull in pounds} = \frac{\text{hp} \times 33{,}000}{\text{Belt speed in feet per min.}}$$

Belt speed in feet per min. = Pulley dia. in inches $\times$ 0.262 $\times$ r.p.m.

For certain classes of service where belted motors operate in dusty locations, as in clay crushing plants, cement mills, flour mills, etc., the action of dust accumulations on the belt tends to reduce its pulling power and necessitate an increase in width of from 25 to 50 per cent over that which would be satisfactory for ordinary conditions. Experience has shown that a four-ply treated canvas or rubber belt is approximately equal in transmitting power to a single or light double leather belt of good quality.

The maximum ratio between driving and driven pulleys should not exceed 6:1 for ordinary conditions, and

the distance between centers will depend on the ratio, good proportions being approximately as follows:

Approx. Ratio.	Min. Distance Between Centers
2:1	8 feet
3:1	10 feet
4:1	12 feet
5:1	15 feet
6:1	20 feet

All belts should have a normal lap on the pulleys of at least 160 degrees unless some form of idler pulley is used to increase belt contact. The use of paper pulleys allows a somewhat smaller degree of lap owing to the better adhesive qualities of this material, and for this reason is to be recommended for all small pulley drives.

It is not usually considered good practice to use heavy double belts of any kind on pulleys smaller than 15 inches diameter, as the abrupt bending of the thick belt over a small pulley soon opens the fibers and ruins the material.

Rope Connection.—On a par with belt driving in its effect on the motor can be placed rope drives of either the English or American system, and many installations are at present using this drive for large motors. The English or separate rope system is a favorite in rolling mill or other work where large powers are transmitted and continuity of operation is especially desirable, while the American or single rope system is especially adapted to group drive from a large unit. For either system as large a rope as possible should be chosen, for economical reasons.

To insure reasonable life of the ropes, the smallest sheave over which the ropes run should be not less than forty-two times the diameter of the rope used. Lubri-

cated core rope should be used, as the principal friction tending to destroy rope is between the fibers themselves and, owing to the action of centrifugal force, external lubricant will not penetrate the rope.

Considering the life of the rope alone, experience shows that the most economical speed is 4,500 feet per minute, and this should be used where cost of maintenance is an important factor.

Very small pulleys or sheaves, in general, and on induction motors in particular, should be avoided, as they necessitate heavy belt tension to transmit the power and invariably produce bearing troubles which militate greatly against efficient operation, and on induction motors with their air-gap clearance cause secondary troubles liable to damage the machine seriously.

Toothed Chain Connection.—In many applications where positive connection is desirable, a toothed chain, such as the Morse or Renolds type, offers a somewhat flexible and a very satisfactory method of motor connection. When the center distances between driving and driven shafts are too short for belts and too great for the use of gears, the maximum ratio obtainable with chain is approximately 6:1. As short a center distance should be used as possible owing to the expense of the chain. For the best operation, the chain speed should not exceed 1,500 feet per minute, as beyond this speed the necessary grease for lubrication is thrown off and the wear becomes excessive. It is also imperative that means be provided for adjusting the sprocket center distances to take up the wear in chain and sprockets, and keep same fairly tight. Chains work best when driving horizontally. The driving side should be on the bottom when distance is long, and on top when distance is short or speed high. For certain very severe conditions spring

center gear sprockets have been used, and found very satisfactory in protecting the motor and increasing the life of the chain. Chain drive imposes a more severe strain upon the revolving parts of a motor than a belt drive does, and this fact must be borne in mind and should be properly discounted in the type and design of motor recommended for a specific application.

Gear and Pinion Connection.—Owing to its economic advantages as to floor space, etc., the trend of new development appears to be in the direction of direct geared applications, and it is where this condition exists that the greatest precautions must be taken to eliminate those undesirable factors which usually result in a rapid depreciation of the motor, unless it has been designed with a view to fitting this particular application.

In general it has been found that where standard motors designed for belted service, have been used in general applications, driving roll trains, bending rolls, plunger pumps, glass grinding machinery, cement mill machinery, clay working machinery, coal and ore handling machinery, hoists, cranes, etc., etc., failure has been very frequent where the mechanical limitations of the electric motor were not properly understood and taken into account.

As an engineering proposition, the use of a standard commercial motor of over 20 horse-power with a pinion mounted directly on the motor shaft should be avoided wherever possible. Where service conditions demand a direct-geared connection, an ideal arrangement, and one to be used wherever possible, is to have the pinion carried on a short shaft in separate pedestals or other bearings; this shaft to be coupled to the motor shaft by some form of flexible or cushion coupling. This arrangement at once removes the motor from the destruc-

tive vibrations and hammering of the gears, allowing a lighter type of motor to be used, and, while somewhat more expensive than the common method of mounting the pinion directly on the motor shaft, the increase in cost is a cheap premium to pay for the practically absolute continuity of operation assured. It is especially applicable in many lines of industrial work where wear on the bearings of the driven mechanism is an unimportant item, and where the motor bearings are liable to be neglected.

Where the above arrangement is not permissible and where operating conditions are not too severe, an outboard bearing for the motor shaft beyond the pinion is to be recommended for motors of 20 horse-power and over, as this auxiliary increases by nearly 100 per cent the resistance of the shaft to bending, and the ability of the bearings to resist pounding out. As the chief cause of trouble on geared motors can be traced directly to the hammering and vibration due to the tooth impacts, the best practice demands that for commercial motors, where the pin is mounted directly on the shaft, speeds should not exceed 700 R.P.M. for motors above 20 horse-power, and for heavy work even this is too high.

The maximum pitch line speed at which cut metal gears should be run is approximately 2,000 feet per minute, and speeds higher than 1,200 feet are not to be recommended for best results, particularly where the pinion is mounted directly on the motor shaft.

Pitch line speed may be determined from the following equation:

P.L.S. = Rev. $\times$ Pitch. Dia. in inches $\times$ 0.262.

$$\text{Pitch Dia. in inches} = \frac{\text{Number of teeth}}{\text{Diametral Pitch}}$$

$$\text{Pitch Dia. in inches} = \frac{\text{Number teeth} \times \text{Circular Pitch}}{3.1416}$$

As in ordinary motor drives the pinions are usually of forged steel, the strength of the gear only need be considered when figuring the size of gear required. Table 37 shows a ready means of determining the proper proportions of cast iron gears having more than 25 teeth to transmit a given horse-power, also the gear pitches preferable for motor drives.

TABLE 37.—SHOWING VALUES OF HP PER INCH WIDTH FOR PITCH LINE SPEEDS ABOVE 1,000 FEET PER MINUTE.*

HP. to be Transmitted	Preferable Diametral Pitch	HP. per Inch Width per 100 ft. per Min.
1 to 2	8	0.24
2 to 5	6	0.30
5 to 10	5	0.38
10 to 30	4	0.48
30 to 75	3	0.64
75 to 125	2½	0.76
125 to 200	1¾	1.10

* For cast steel gears the values in HP per inch width should be increased 2½ times.

For pitch line speeds from 500 to 1,000 feet per minute the above values of horse-power should be increased 50 per cent, and for speeds up to 500 feet per minute should be increased 100 per cent.

The minimum pitch diameter of motor pinion is approximately twice the diameter of shaft for ordinary cases, and the smallest pinion should not have less than 14 teeth under any condition.

Example:—25 horse-power motor, 600 R.P.M., countershaft speed 100 R.P.M.

Diameter motor shaft at pinion fit $= 2\frac{1}{2}$ inches.

Smallest pitch diameter possible $= 2 \times 2\frac{1}{2} = 5$ inches.

From table of preferable pitches, pitch $= 4$ inches.

Therefore, smallest number of teeth $= 4 \times 5 = 20$.

Pitch line speed $= 600 \times 5 \times 0.262 = 786$ feet per minute.

$$\text{Horse-power per 100 feet per min.} = \frac{25 \times 100}{786} = 3.18.$$

From the table it is evident that horse-power per inch of width at this speed $= 0.48 +$ (50 per cent) $= 0.72$.

$$\text{Therefore, the proper face of cast iron gear} = \frac{3.18}{0.72}$$

$= 4.41$, or, say, $4\frac{1}{2}$ inches.

$$\text{Proper number of teeth for gear} = \frac{600 \times 20}{100} = 120$$

teeth.

Where cast iron gear faces figure out greater than seven inches, it will be economical to use cast steel gears, as by their use the face may be made one-half the size.

When high speed motors must be used in geared-applications, experience has shown that rawhide pinions

alleviate vibrations, noise, etc., to some extent, although not sufficiently to make unnecessary such precautions as should be taken where steel pinions are used. The value of the rawhide lies principally in its noiselessness of operation, especially where the rawhide is not shrouded, viz., where the design is such that the retaining brass checks are cut below the roots of the teeth. With such forms of pinion, pitch line speeds of 3,000 feet per minute have been used successfully on motors up to 50 horsepower. The strength of rawhide is approximately the same as cast iron, and this fact must be borne in mind when considering the use of rawhide pinions, as it will be evident that a steel gear cannot be driven to its full capacity by a rawhide pinion.

For a majority of motor applications, involute forms of cut gear teeth are to be recommended, as this form does not require the maintenance of exact center distances demanded by other forms of teeth. Also, because this form has been adopted as standard by the majority of gear manufacturers under the diametral pitch system.

Cushion Couplings.—In many applications, such as air compressors, plunger pumps, etc., the motor is required to be coupled directly to an intermediate shaft which usually carries a pinion for transmitting power to other parts of the mechanism. Application of a motor to this service should be through a cushion coupling for best results, as with the usual forms of rigid coupling the shocks and vibrations from the gears due to the cyclical variations in the load are transmitted back to the motor and are very destructive to the revolving windings. Vertical motors driving centrifugal pumps through long vertical shafts should always be connected together by means of a flexible coupling that will allow

considerable displacement in shaft alignment as, owing to the absence of gravitational forces in the plane of revolution, all vertical running motors and shafts are extremely sensitive and destructive vibrations are easily started.

Flange Couplings.—For installations where the motors have to deliver a practically constant torque, and where there is little liability of shafts getting out of alignment, such as centrifugal pumps, blowers, motor-generator sets, etc., a flange or rigid form of coupling has been found entirely satisfactory. A careful investigation should always be made, however, of all factors liable to influence operation, as successful application of this form of coupling requires practically ideal conditions.

Foundations.—In many motor installations insufficient attention and care are bestowed on the motor foundation, or method of mounting to insure satisfactory motor operation. This is particularly true of geared applications where motors are frequently found shimmed up with wooden blocks. This should not be tolerated, as it can be nothing else than unsatisfactory; rigidity of mounting or foundation being of extreme importance in geared applications.

PIPES AND PIPE FITTING.

As has been previously stated, it frequently happens that the millwright is called upon to superintend the laying out and erection of the steam and water piping systems for a mill or factory, and considering this to be the case, a space will be devoted to the discussion of this most important subject.

Selection of Pipe.—This is an important matter, and

care and good judgment should be exercised in the choice of pipe that is to be used on the job. The bulk of the pipe manufactured today is steel, it having been demonstrated by experiment and also by practice that steel pipe is far superior to iron pipe for durability, and hard usage, consequently it has come to stay. Therefore one of the requirements of the successful pipe fitter is to be able to distinguish steel pipe from iron pipe, and a few pointers on this subject will be presented.

In the first place, iron pipe is rough in appearance and the scale on it is heavy, whereas on steel the scale is very light, and has the appearance of small blisters or bubbles, underneath which the surface is smooth and somewhat white. When flattened, steel pipe seldom breaks; but if a fracture does occur, it will be noticed that the grain is very fine. Iron pipe, when subjected to this test, breaks readily and shows a coarse fracture, due to the long fiber of this material.

The impression often prevails that steel pipe is exceedingly hard, for which reason they imagine that it is threaded with difficulty and that the threads are easily broken off. This belief is entirely erroneous, the truth being that steel pipe is soft and tough. Threads on this pipe do not break; they tear off, to avoid which it is necessary that the cutting die shall be sharp and thus cut above the center. Dies suitable for steel pipe can also be used on iron pipe; but blunt dies that will work successfully on iron pipe will tear the threads on steel pipe, owing to the softness of the metal.

Laying Out a System of Piping.—A good rule to follow in laying out a piping system is as follows: If for steam, start at the distributing end of the system, and work toward the boilers, gradually increasing the size of the pipe at each outlet until all have been provided for,

and the total area of which should not exceed the area of the steam main leading from the boilers. For water the same rule will apply, always bearing in mind that, at the same velocity of flow the capacities of pipes are proportional to the squares of their internal diameters, as for instance one 4-inch pipe will deliver the same volume as four 2-inch pipes. With the same head or pressure, however, the velocity is less in the smaller pipe, and the volume delivered varies about as the square root of the fifth power (*i. e.,* as the 2.5 power).

Table 38 will be found very convenient in calculating sizes of pipes, and the number of smaller sized pipes required to equal in capacity one larger.

The method of using the table is as follows: Assume the main or header leading through the factory to be a 4-inch pipe. Reference to Table 38 shows that the internal area of a 4-inch pipe equals 12.730 square inches. From this header the following branches may be safely led:

One	2-in. pipe, internal area =	3.356 sq. in.
Two	1½-in. pipes, internal area =	4.072 sq. in.
Two	1¼-in. pipes, internal area =	2.992 sq. in.
Two	1-in. pipes, internal area =	1.722 sq. in.
	Total combined area =	12.142 sq. in.

and 12.730 — 12.142 = .588 sq. in. as a margin.

For the smaller sizes of branch pipes, say from 1 in. down to ½ in., which are likely to be tapped into the above mentioned pipes, the table may again be consulted, as for instance, from the 2-in. pipe having an area of 3.356 sq. in., there may be taken:

One 1¼-in. pipe, internal area = 1.496 sq. in.
Two ¾-in. pipes, internal area = 1.066 sq. in.
Two ½-in. pipes, internal area = .608 sq. in.

Total combined area = 3.170 sq. in.

and 3.356 — 3.170 = .186 sq. in. as a margin.

Table 38 follows:

TABLE 38.—INTERNAL AREA OF WROUGHT IRON OR STEEL WELDED PIPE.

Nominal Inside Diameter Inches	Internal Area Sq. Inches	Nominal Inside Diameter Inches	Internal Area Sq. Inches
⅜	.191	5	19.986
½	.304	6	28.890
¾	.533	7	38.738
1	.861	8	50.027
1¼	1.496	9	62.730
1½	2.036	10	78.823
2	3.356	11	95.033
2½	4.780	12	113.098
3	7.383	13	137.887
3½	9.887	14	159.485
4	12.730	15	182.665
4½	15.961		

Ordinary or standard black pipe, from ⅛ to 1 inch inclusive, is butt welded, and tested to 300 pounds to the square inch. Pipe 1¼ inches and larger is lap welded, and proved to 500 pounds to the inch, and the millwright should keep well within these figures when handling steam or water under heavy pressures. There is also made and sold from stock a heavier pipe known as "hydraulic." This pipe should be used when very heavy pressures are to be resisted, particularly when there are shocks similar to those when the hydraulic ram is used.

Expansion and Contraction.—In laying out a system of steam-piping for a power plant, perfect freedom for expansion and contraction should be allowed, to prevent

undue strains on any member of the system, or at the joints, causing them to leak. The old types of slip-expansion joints having proved a constant source of trouble and expense requiring frequent repacking and adjusting, are seldom, if ever, used on a good job of piping. If absolutely necessary, however, to use this type of expansion joint, the piping should be so anchored as to prevent the joint from pulling apart.

With the advent of higher steam pressures and correspondingly higher temperatures and velocities, more attention has been given to the proper designing of piping systems, and constructive details than in the past. Steel

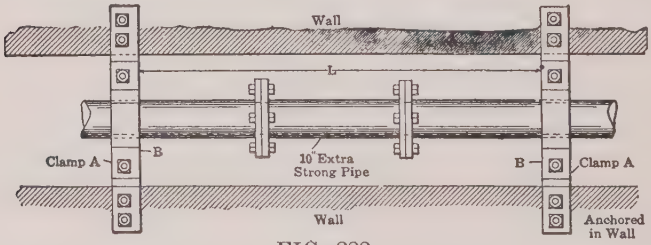


FIG. 222.

pipe bends of long radius are used wherever practical in place of the cast elbows of short radius. They take up the expansion stresses, making the system more flexible throughout; reduce vibrations and friction, and deliver the steam to the engine with a lower drop in pressure.

Pipe bends curved to a radius of less than five diameters of the pipe are undesirable as expansion bends. The radius should be at least five or six pipe diameters.

Anchoring.—This is an important detail, especially in large sized pipe, and in order to be able to correctly locate the anchors, the designer should possess a good knowledge of the expansion of pipes when heated, and

their contraction when cold. Fig. 222 shows a 10-inch extra heavy steam pipe, and two clamps, A A, anchored rigidly to the wall. Assume the pipe to have been placed in position while at a temperature of 60° Fahr. and held securely in one clamp, and allowed to move freely through the other one while expanding. Now if while in this condition, steam at 250 lbs. pressure, and superheated 100 degrees is turned into the pipe, the elongation of the pipe due to expansion will be 1⅞ in. in each 60 feet of length; that is, the pipe which was originally 60 feet long, is now 60 feet 1⅞ inches in length, due to the added temperature of 441 degrees, and the movement of the pipe through the loose clamp is 1⅞ inches. But suppose the pipe to have been rigidly clamped between both clamps, with no chance for endwise movement during the process of heating, the result would be that either the clamps would spring, or the pipe would buckle, or bend sideways, owing to its great length as compared with its diameter. The tremendous force exerted upon the material, and joints of the pipe by the thrust of expansion, and the pull of contraction may be calculated by means of the following formula: P = C T A E in which P = thrust in lbs. from expansion, or pull due to contraction. C = co-efficient of linear expansion = 0.000006. T = difference in degrees Fahr. of temperature between original temperature and temperature after pipe has been heated, or cooled = 441 degrees. A = area of metal in cross section of pipe in square inches which for 10-inch extra heavy pipe = 16. E = co-efficient of elasticity = 30,000,000. By substituting these values we have: P = 0.000006 × 441 × 16 × 30,000,000 = 1,270,080 pounds.

These figures give some idea of the strains to which fittings and joints are subject where expansion and contraction are not properly provided for in steam lines,

or where anchors are improperly located. As another illustration of the strains exerted upon the fiber of the pipe, and upon joints and fittings, suppose that the pipe already referred after being heated and allowed to expand freely, is then rigidly clamped at both ends, and allowed to cool off again to its original temperature, the tensile strain upon the material in the pipe would be:

$$\frac{P}{A} = \frac{1{,}270{,}080}{16} = 80{,}000 \text{ lbs. per square inch,}$$

which would cause rupture.

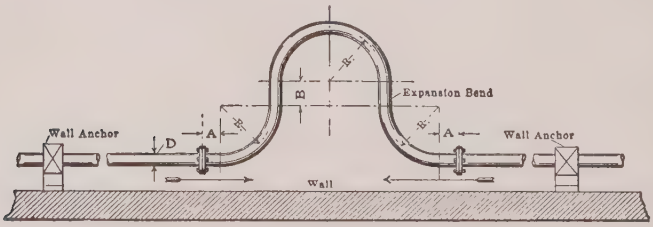


FIG. 223.
FOR LONG RUNS.

Allowing for Expansion and Contraction.—It is considered good practice in figuring for expansion to allow only one-half the calculated amount when cutting the pipe to length. For example, if in a run of pipe 100 feet between connections, or points where steam lines are taken off from the header, the expansion is calculated to be 3 inches; allow 1½ inches when cutting the pipe to length, or in other words, the total length of pipe should be 100 — 1½ = 99 feet 10½ inches.

Then the steam fitter takes up the other 1½ inches when erecting the line, with the result that when steam is turned on the expansion removes the tension or strain

put on the pipe when cold, and the fittings and joints are strained only one-half as much as if none, or all of the expansion were allowed for. This rule is used by most large concerns, and has proven to be satisfactory to operating conditions in general.

The expansion bend shown in Fig. 223 is intended for use in a long run of piping where it is desirable to anchor the line at two points to take the strain off the fittings and joints outside the anchors, not shown. When heated, the pipe expands in the direction of the arrows,

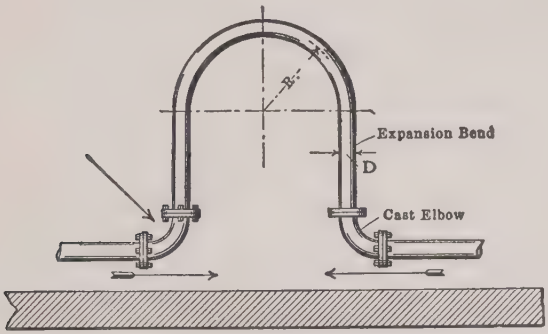


FIG. 224.

tending to close the bend. To insure sufficient elasticity the dimension B should be made as large as possible. The dimensions A and R can be respectively 5 and 1 of D.

For sizes above 6 inches it may be impossible to make this bend in one piece, owing to the long length of pipe required. Commercial pipe averages from 16 to 20 feet in length, although lengths up to 24 feet are kept in stock by some dealers for use in making up special bends.

For bends 8 inches in diameter and larger, the arrangement shown in Fig. 224 may be used for the same

purpose. With such an arrangement, however, a bending strain is put on the joints, as shown in Fig. 225, which is greatly enlarged to show the straining action on the joint at E, as the pipe expands.

The cast elbows should be extra heavy, with thick flanges. The flanges on the bend should be thick also, as a thin flange is easily ruptured or strained sufficient to cause leakage at the joints.

Where there is sufficient room, a better arrangement would be to substitute two 90-degree or square bends of steel pipe in place of the cast elbows, making the system more flexible.

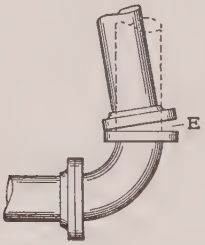


FIG. 225.

One method of anchoring a steam header of moderate length is shown in Fig. 226. The line of pipe which may be from 50 to 150 feet in length is securely anchored at the middle of its length, and allowed to expand in both directions, as shown by the arrows A A. The joints nearest the anchor or clamp suffer the least strain, while those joints nearer the ends of the header are subject to the greatest strains from expansion and contraction. In laying out a system of piping, the amount of expansion for the different runs should be worked out, and provision made accordingly. The expansion of cast iron in fittings is about the same as untempered steel,

but under the strain and stress of superheated steam (combined heat and pressure), a fitting 22 inches in length from face to face of flanges has been known to have a permanent elongation of $\frac{5}{16}$ in. in four years' use. Therefore, in the design of steam pipe lines, the question of superheat must be considered as tending toward the permanent elongation, and consequent loss of tensile strength of the material.

Fittings and Valves.—For flanged fittings there are two recognized standards: one for pressures up to 125 pounds, adopted by the A. S. M. E., and another for pressures up to 250 pounds, adopted by the manufactur-

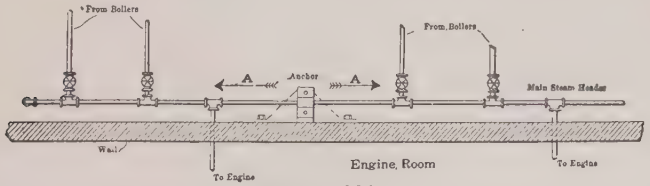


FIG. 226.

ers. Some manufacturers list their pipe and fittings as "standard" for pressures up to 125 lbs., "full weight" for pressures from 125 lbs. to 175 lbs., and "extra heavy" for pressures higher than 175 lbs.

Flanged valves are made with the thickness of flange to correspond with thickness of pipe flange, and the "standard," "medium," or "extra heavy" correspond with the same grades of pipe and pipe flanges.

It is a generally accepted fact that few, if any, of the accidents caused by the failure of pipe fittings are due only from the pressure of the gas or fluid contained in them, but are the result of combined stresses caused by the expansion and contraction of the pipe, water hammer, or pipe improperly supported, any or all of these

being in conjunction with a stress due to the pressure of the gas or fluid within the fitting.

A fitting may, therefore, be subjected to two classes of stresses: one, a legitimate stress, due to the pressure from that which the fitting contains, such as water, steam, compressed air, etc., and the other, due to strains caused by systems of piping improperly designed or installed. Stresses due to the former are tangible and may be estimated with some degree of accuracy, but stresses due to the latter are an unknown quantity varying with the wisdom of the designer and for which the factor of safety must provide.

The dimensions of fittings of different makers vary slightly, so that what is true of certain sizes of one maker's fittings might not be true of another's.

Table 39 gives the bursting strength of standard cast iron fittings, screwed. It will be seen that the elbows show a greater strength for all sizes than do the tees. Table 40 gives the ultimate test pressure of malleable iron pipe fittings including both black, and galvanized. The bodies of these fittings were .25 in. in thickness, and the average bursting pressure of the black fittings was 2,900 lbs., while that of the galvanized fittings was 2,833 lbs. These important details concerning pipe fittings were obtained through a series of tests conducted by Mr. S. M. Chandler at the laboratories of a prominent valve and fittings manufacturing company.

TABLE 39.—STANDARD SCREWED CAST-IRON ELBOWS AND TEES—ULTIMATE TEST PRESSURES.

Size	Elbows		Average	
2¼	3500	3300	3400	3400
3	2400	2600	2100	2500
3½	2100	1700	2400	2250
4	2800	2500	2500	2600
4½	2000	2600	2600	2600
5	2600	2500	2500	2533
6	2600	2200	2300	2367
7	1800	2100	1900	1950
8	1700	1600	1700	1667
9	1800	1800	1900	1833
10	1800	1700	1600	1700
12	1100	1200	900	1150

Size	Tees		Average	
1¼	3400	3300	3300	3333
1½	3400	3200	2800	3300
2	2500	2800	2500	2600
2½	2400	2100	2500	2450
3	1400	1900	1800	1850
3½	1200	1500	1800	1650
4	1800	2100	1700	1867
4½	1100	1400	1400	1400
5	1700	1300	1500	1600
6	1400	1500	1100	1450
7	1400	1400	1500	1433
8	1200	1400	1300	1350
9	1300	1400	1200	1300
10	1100	1300	1200	1200
12	1100	1000	1100	1067

TABLE 40.—BURSTING PRESSURE OF MALLEABLE PIPE FITTINGS.

Test No.	Black or Galvanized	Ultimate Test Pressure	Weight, Pounds
1	Black	2800 lb.	10.06
2	Black	3100 lb.	10.12
3	Black	2800 lb.	10.19
4	Galvanized	2700 lb.	10.62
5	Galvanized	3000 lb.	10.50
6	Galvanized	2800 lb.	10.75

Testing for Leaks.—When the system is completed, a test should be made for leaks, particularly obscure leaks. Leaks in water piping will quickly manifest themselves when the pressure is turned on, but for steam pipes some little time will be required for the steam to condense in the pipes, when the first intimation of a leak will be water dripping from it. Very often there are steam leaks which, owing to the dryness of the steam, may be heard but not easily located. A good way to search for leaks is to prepare a pail full of strong soap-suds, and using a paint brush go over the lines of pipe, applying the mixture to each one of the joints. If there is even a very slight leak any place where the soap-suds touch, bubbles will be blown, and the leak located. It should then be marked with a piece of white chalk carried along for the purpose.

Air and Water Traps.—When running pipe lines for steam, the great thing to be avoided is the forming of pockets in which water may collect and thus cut down

the capacity of the pipe. Under certain conditions, the pipe may become completely filled with water in spots and the flow of steam cut off until a lowering of pressure at the discharge end is sufficient to overcome the hydrostatic pressure of the collected water and that substance is driven out of the pipe. Water traps or "pockets" in a steam pipe are always to be avoided. They are very undesirable, and in some instances they become dangerous.

Water pipes are also cut down in capacity by pockets which may contain air. But in this case, instead of a low place in the pipe giving trouble, the high spots are to be avoided. A collection of air in the bend of a pipe passing over a high place in the line may, in many instances, cut down the pressure greatly, and in some cases entirely stop the flow of water. Particularly is this the case when the pipe acts as a syphon; the pressure or weight of the water in the discharge leg must overcome the weight of water in the supply leg of the syphon.

In water pipes under pressure, the action is the same, though the force not being limited to 14.7 pounds to the square inch can overcome to better advantage the presence of air in the pipe. In either case, the loss of pressure amounts to the loss of head equal to the height of column of air in the pipe, calculated for an equal height of water. Thus, pockets and traps must be avoided in both steam and water piping, and the straighter the pipes can be run, in a vertical direction, the better will the pipe line serve its purpose.

Dead-Ends and Drips.—Dead-ends in pipe lines should always be avoided if possible. No matter where placed, a dead-end always fills with water. If the pipe be a large one and the dead-end projects vertically upward, the water may run out as fast as it collects, but

unless a dead-end is drained in some way, it will surely fill with water—and stay filled. The manner of its filling is through the condensation of the steam which flows to the dead-end. A pipe with a close end is much like an extended bellows. The condensing of the steam in the pipe is like the closing of the bellows. As everything—boards and leather—is drawn down upon the bottom board of the bellows, so every bit of condensed water is drawn to the dead-end of a steam pipe by the condensing of the steam in that portion of the pipe.

When it is required to maintain a circulation in a dead-end, a drip should be provided which will take care of the water of condensation as fast as it collects. Either a drip must be established or a loop must be made, whereby what would be the dead-end is connected to some portion of the steam system by means of which the water of condensation may pass away. Hand operated drips may be used, but this is at the best, a wasteful and unreliable method of disposing of the condensation.

A much better method is the employment of automatic traps, discharging into a return main which will carry the water of condensation back to a receiver in the boiler room or pump room, from whence it may be taken by the feed pump and returned to the boilers.

Making Up Pipe Joints.—For making up permanent screw joints in pipe, where it is not necessary to break the joints again, nothing is better than good red lead, though litharge (yellow oxide of lead) answers nearly as well. For screw joints which may have to be taken down frequently, plumbago mixed with cylinder oil, makes a good material for daubing either threads or flanges; it is also good on flange bolt threads. For threaded pipe, use as thick as possible. A mixture which will run is too thin. It daubs things too much. For

flange use, it may be mixed thin enough to allow of being spread on the surfaces with a brush. When nice work is required, daub the inside of fittings, valves, etc., and it will not show as when daubed on pipe, nipples, etc.

USEFUL INFORMATION.

One heaped bushel of anthracite coal weighs from 75 to 80 lbs.

One heaped bushel of bituminous coal weighs from 70 to 75 lbs.

One bushel of coke weighs 32 lbs.

Water, gas and steam pipes are measured on the inside.

One cubic inch of water evaporated at atmospheric pressure makes 1 cubic foot of steam.

A heat unit known as a British Thermal Unit raises the temperature of 1 pound of water 1° Fahr.

For low pressure heating purposes, from 3 to 8 pounds of coal per hour is considered economical consumption, for each square foot of grate surface in a boiler, dependent upon conditions.

A horsepower is estimated equal to 75 to 100 square feet of direct radiation. A horsepower is also estimated as 15 square feet of heating surface in a standard tubular boiler.

Water boils in a vacuum at 98° Fahr.

A cubic foot of water weighs 62½ pounds, it contains 1,728 cubic inches or 7½ gallons. Water expands in boiling about one-twentieth of its bulk.

In turning into steam water expands 1,700 times its bulk, approximately 1 cubic foot of water will produce 1 cubic foot of steam.

One pound of air contains 13.82 cubic feet.

It requires 1½ British Thermal Units to raise one cubic foot of air from zero to 70° Fahr.

At atmospheric pressure 966 heat units are required to evaporate one pound of water into steam.

A pound of anthracite coal contains 14,500 heat units.

One horsepower is equivalent to 42.75 heat units per minute.

One horsepower is required to raise 33,000 pounds one foot high in one minute.

To produce one horsepower requires the evaporation of 2.66 pounds of water.

One ton of anthracite coal contains about 40 cubic feet.

One bushel of anthracite coal weighs about 86 pounds.

Heated air and water rise because their particles are more expanded, and therefore lighter than the colder particles.

A vacuum is a portion of space from which the air has been entirely exhausted.

Evaporation is the slow passage of a liquid into the form of vapor.

Increase of temperature, increased exposure of surface, and the passage of air currents over the surface, cause increased evaporation.

Condensation is the passage of a vapor into the liquid state, and is the reverse of evaporation.

Pressure exerted upon a liquid is transmitted undiminished in all directions, and acts with the same force on all surfaces, and at right angles to those surfaces.

The pressure at each level of a liquid is proportional to its depth.

With different liquids and the same depth, pressure is proportional to the density of the liquid.

The pressure is the same at all points on any given level of a liquid.

The pressure of the upper layers of a body of liquid on the lower layers causes the latter to exert an equal reactive upward force. This force is called buoyancy.

Friction does not depend in the least on the pressure of the liquid upon the surface over which it is flowing.

Friction is proportional to the area of the surface.

At a low velocity friction increases with the velocity of the liquid.

Friction increases with the roughness of the surface.

Friction increases with the density of the liquid.

Friction is greater, comparatively, in small pipes, for a greater proportion of the water comes in contact with the sides of the pipe than in the case of the large pipe. For this reason mains on heating apparatus should be generous in size.

Air is extremely compressible, while water is almost incompressible.

Water is composed of two parts of hydrogen, and one part of oxygen.

Water will absorb gases, and to the greatest extent when the pressure of the gas upon the water is greatest, and when the temperature is the lowest, for the elastic force of gas is then less.

Air is composed of about one-fifth oxygen and four-fifths nitrogen, with a small amount of carbonic acid gas.

To reduce Centigrade temperatures to Fahrenheit, multiply the Centigrade degrees by 9, divide the result by 5, and add 32.

To reduce Fahrenheit temperature to Centigrade, subtract 32 from the Fahrenheit degrees, multiply by 5 and divide by 9.

To find the area of a required pipe, when the volume and velocity of the water are given, multiply the number of cubic feet of water by 144 and divide this amount by the velocity in feet per minute.

Water boils in an open vessel (atmospheric pressure at sea level) at 212° Fahr.

Water expands in heating from 39 to 212° Fahr., about 4 per cent.

Water expands about one-tenth its bulk by freezing solid.

Rule for finding the size of a pipe necessary to fill a number of smaller pipes.—Suppose it is desired to fill from one pipe, a 2, 2½ and 4-inch pipe. Draw a right angle, one arm 2 inches in length, the other 2½ inches in length. From the extreme ends of the two arms draw a line. The length of this line in inches will give the size of pipe necessary to fill the two smaller pipes—about 3¼ inches. From one end of this last line, draw another line at right angles to it, 4 inches in length. Now, from the end of the 2-inch line to the end of the last line draw another line. Its length will represent the size of pipe necessary to fill a 2, 2½ and 4-inch pipe. This may be continued as long as desired.

Discharge of Water.—The amount of water discharged through a given orifice during a given length of time and under different heads, is as the square roots of the corresponding heights of the water in the reservoir above the surface of the orifice.

Water is at its greatest density and occupies the least space at 39° Fahr.

Water is the best known absorbent of heat, consequently a good vehicle for conveying and transmitting heat.

A U. S. gallon of water contains 231 cubic inches and weighs 8 1/3 pounds.

A column of water 27.67 inches high has a pressure of 1 pound to the square inch at the bottom.

Doubling the diameter of a pipe increases its capacity four times.

A cubic foot of anthracite coal averages 50 pounds. A cubic foot of bituminous coal weighs 40 pounds.

Weights.

One cubic inch of water weighs....	0.036	pounds
One U. S. gallon weighs.....	8.33	"
One Imperial gallon weighs	10.00	"
One U. S. gallon equals.....	231.00	cubic inches
One Imperial gallon equals.....	277.274	" "
One cubic foot of water equals.....	7.48	U. S. gallons

Liquid Measure.

4 Gills make 1 Pint
2 Pints make 1 Quart
4 Quarts make 1 Gallon
31½ Gals. make 1 Barrel

To find the area of a rectangle, multiply the length by the breadth.

To find the area of triangle, multiply the base by one-half the perpendicular height.

To find the circumference of a circle, multiply the diameter by 3.1416.

To find the area of a circle, multiply the diameter by itself, and the result by .7854.

To find the diameter of a circle of a given area, divide the area by .7854, and find the square root of the result.

To find the diameter of a circle which shall have the same area as a given square, multiply one side of the square by 1.128.

To find the number of gallons in a cylindrical tank, multiply the diameter in inches by itself, this by the height in inches, and the result by .34. To find the number of gallons in a rectangular tank, multiply together the length, breadth and height in feet, and this result by 7.4. If the dimensions are in inches, multiply the product by .004329. To find the pressure in pounds per square inch, of a column of water, multiply the height of the column in feet by .434.

To find the head which will produce a given velocity of water through a pipe of a given diameter and length: Multiply the square of the velocity, expressed in feet per second, by the length of pipe multiplied by the quotient obtained by dividing 13.9 by the diameter of the pipe in inches, and divide the result obtained by 2,500. The final amount will give the head in feet.

Example.—The horizontal length of pipe is 1,200 feet, and the diameter is 4 inches. What head must be secured to produce a flow of 3 feet per second?

$$3 \times 3 = 9;\ 13.9 \div 4 = 3.475.$$
$$9 \times 1{,}200 \times 3.475 = 37{,}530.$$
$$37{,}530 \div 2{,}500 = 15 \text{ ft.}$$

To find the velocity of water flowing through a horizontal straight pipe of given length and diameter, the head of water above the center of the pipe being known: Multiply the head in feet by 2,500, and divide the result by the length of pipe in feet multiplied by 13.9, divided by the inner diameter of the pipe in inches. The square root of the quotient gives the velocity in feet per second.

To find the head in feet, the pressure being known, multiply the pressure per square inch by 2.31.

To find the contents of a barrel.—To twice the square of the largest diameter, add the square of the smallest

diameter and multiply this by the height, and the result by 2,618. This will give the cubic inches in the barrel, and this divided by 231 will give the number of gallons.

To find the lateral pressure of water upon the side of a tank, multiply in inches, the area of the submerged side, by the pressure due to one-half the depth.

Example.—Suppose a tank to be 12 feet long and 12 feet deep. Find the pressure on the side of the tank.

$144 \times 144 = 20{,}736$ square inches area of side.

$12 \times .43 = 5.16$, pressure at bottom of tank. Pressure at the top of tank is 0. Average pressure will then be 2.6. Therefore $20{,}736 \times 2.6 = 53{,}914$ pounds pressure on side of tank.

To find the number of gallons in a foot of pipe of any given diameter, multiply the square of diameter of the pipe in inches, by .0408.

To find the diameter of pipe to discharge a given volume of water per minute in cubic feet, multiply the square of the quantity in cubic feet per minute by 96. This will give the diameter in inches.

To find the weight of any length of lead pipe, when the diameter and thickness of the lead are known: Multiply the square of the outer diameter in inches, by the weight of 12 cylindrical inches, then multiply the square of the inner diameter in inches by the same amount, subtracting the product of the latter from that of the former. The remainder multiplied by the length gives the desired result.

Example.—Find the weight of 1,200 feet of lead pipe, the outer diameter being $\frac{7}{8}$ inch, and the inner diameter 9/16 inch.

The weight of 12 cylindrical inches, 1 foot long, 1 inch in diameter, is 3.8697 lbs.

$7/8 \times 7/8 = 49\text{-}64 = .765625.$
$9/16 \times 9/16 = 81\text{-}256 = .316406.$
$.765625 - .316406 = .449219 \times 3.8697 \times 1{,}200 =$ 2,086 lbs.

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